Universidade de Aveiro Departamento Engenharia Mecânica
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Conceção de bomba de óleo de caudal ajustável para motores de combustão interna

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Develoment of Variable Displacement Oil Pump for Internal Combustion Engines

Dissertação apresentada à Universidade de Aveiro para cumprimento dos requisitos necessários à obtenção do grau de Mestre em Engenharia Mecânica, realizada sob a orientação científica do Professor Doutor António Completo e Professor Doutor Fernando Neto, Professores do Departamento de Engenharia Mecânica da Universidade de Aveiro.

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Palavras-chave

Bomba de óleo, Bomba de óleo automóvel, Bomba de óleo de débito variável, Bomba de óleo de caudal ajustável, Bomba de óleo regulável, Desenvolvimento de Produto.

Resumo

Nos últimos anos, a necessidade de desenvolver carros amigos do ambiente e mais eficientes levou ao desenvolvimento de várias tecnologias para melhorar o desempenho de motores de combustão interna, uma grande parte das inovações são focadas nos sistemas auxiliares do motor, incluindo a bomba de óleo, sendo este um elemento de grande importância para a dinâmica do motor, bem como um considerável consumidor de energia. Maioria das soluções até aos dias de hoje para bombas de óleo são de caudal fixo, para velocidades médias e altas, o fluxo da bomba é superior às necessidades do motor, este excesso de fluxo leva à necessidade de recirculação do fluido que o que representa um desperdício de energia. Recentemente, os avanços tecnológicos nesta área têm levado à criação de bombas de óleo de caudal ajustável, estas têm-se tornado um 'tem de ter' devido às inúmeras vantagens que trazem, embora o princípio de funcionamento das bombas com palhetas ou pistões sejam relativamente bem conhecidos, a aplicação desta tecnologia para a indústria automóvel é recente e traz novos desafios.

O foco desta dissertação está no desenvolvimento de um novo conceito de bomba de óleo de caudal ajustável para o sector automóvel. O principal objectivo é a obtenção de um conceito que, sendo totalmente adaptável às soluções existentes no mercado (motores), tanto dimensionalmente como nas especificações de desempenho, possua um sistema mecânico inovador para a obtenção de caudal ajustável.
A bomba desenvolvida é um sistema de palhetas com caudal ajustável indo em linha com as soluções comercias existentes, no entanto, a variação da excentricidade habitualmente utilizada a fim de proporcionar um caudal ajustável é anulada, o débito variável é obtido sem variação da excentricidade do sistema, mas com uma variação do comprimento da camara de bombagem. O princípio de funcionamento é diferenciado às soluções, actuais mantendo no entanto a capacidade de integrar peças padrão como válvulas de controlo mecânico e válvulas de segurança, a bomba é compatível com as soluções comerciais em termos de interfaces para ligação entre os sistemas de motor e bomba. Foi desenvolvido um protótipo conceptual do produto de modo a melhor avaliar a validade do conceito.

O conceito desenvolvido representa uma inovação em *design* de bombas de óleo, sendo única no seu sistema mecânico para entrega de caudal ajustável.

Key-Words

Oil Pump, Vehicle oil pump, Variable displacement oil pump, Adjustable flow oil pump, Variable Oil Pump, Product Development.

Abstract

In the last years the need to develop more environmentally friendly and efficient cars as led to the development of several technologies to improve the performance of internal combustion engines, a large part of the innovations are focused in the auxiliary systems of the engine, including, the oil pump, this is an element of great importance in the dynamics of the engine as well a considerable energy consumer. Most solutions for oil pumps to this day are fixed displacement, for medium and high speeds, the pump flow rate is higher than the needs of the engine, this excess flow leads to the need for recirculation of the fluid which represents a waste of energy. Recently, technological advances in this area have led to the creation of variable displacement oil pumps, these have become a 'must have' due to the numerous advantages they bring, although the working principle of vane or piston pumps is relatively well known, the application of this technology for the automotive industry is new and brings new challenges.

The focus of this dissertation is to develop a new concept of variable displacement system for automotive oil pumps. The main objective is to obtain a concept that is totally adaptable to existing solutions on the market (engines), both dimensionally as in performance specifications, having at the same time an innovative mechanical system for obtaining variable displacement. The developed design is a vane pump with variable displacement going in line with existing commercial solutions, however, the variation of the eccentricity commonly used to provide an variable displacement delivery is not used, the variable displacement is achieved without varying the eccentricity of the system but with a variation of the length of the pumping chamber. The principle of operation of the pump is different to existing solutions while maintaining the ability to integrate standard parts such as control valves and mechanical safety valves, the pump is compatible with commercial solutions in terms of interfaces for connection between engine systems and pump. A concept prototype of the product was obtained in order to better evaluate the validity of the concept.

The developed concept represents an innovation in oil pumps design, being unique in its mechanical system for variable displacement delivery.

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Chapter 1

Introduction

1.1 Motivation

In the last few years a run for green and more efficient cars has led to the development of several technologies to improve the internal combustion engine performance. The technologies innovations are largely focused in the improvement of the engine peripheral systems once these are power consumption elements. One of the elements that have suffered changes due to investigations and new solutions development is the central oil pump. This element is responsible for the engine temperature control and elements lubrication.

Currently most of the heat engines of automotive uses fixed-displacement oil pumps oil. These oil pumps are robust and reliable. However, being fixed-displacement (the oil is pumped into the space between gear teeth which has a fixed volume) these does not adapt to different engine speeds and flow needs. For medium and high speeds, the pump flow is greater than the needs of the engine. This excess flow would cause problems in some components. To avoid these problems, fixed flow rate pumps have a recirculation/discharge valve, this recirculation (oil not pumped into the engine system), represents a power/energy loss. One of the strategic priorities in the evolution of internal combustion engines is precisely trying to reduce as much as possible energy losses in order to obtain lower fuel consumption and reduced CO_2 emissions.

Some builders have recently started to produce oil pumps with variable displacement capacity, which is becoming a trend in the market, since it has several advantages for the vehicle performance. These pumps use manly the technology of movable vanes. The variation of displacement is achieved by moving the camera relative to the axis of the rotor where the vanes are housed (eccentricity variation).

However, although the operating principle of the pumps with vanes or pistons being relatively well known, the application of this technology to the automotive industry is recent and brings new challenges. This creates an opportunity for the creation of new and different solutions for this market.

1.2 Objectives

Between the challenges in the development of new solutions is often the differentiation/innovation regarding a panoply of protections performed by builders and suppliers through patents, which always requires a high level of innovation in order to evade the protections already performed. The second main challenge is with the reliability of these pumps, as a result of having more movable parts entails greater mechanical complexity and thus higher risk of reliability, which in critical component it is of the outmost importance. Another challenge is with the fact that these pumps, which have more components than conventional fixed-displacement pumps, these also have to be mounted in an limited space and be within the acceptable range of weight values, otherwise the advantages are annulled. Other challenge relates to the system of command and control (displacement) of the compression/pumping chamber of the pump in order to be reliable, low power consumption and be perfectly adaptable to the needs of engine operation every instant is mandatory. The system of command and control can be mechanical, hydraulic and electro-hydraulic, for example it is possible to get information about the conditions of engine operation through the electronic unit (ECU) of the same. Subsequently, the engine ECU regulates the pressure and flow of oil, depending on the engine speed and load conditions. The electro-hydraulic version has a good cost/benefit and represents the ‘state-of-the-art ' of tomorrow. There is no provision for the future. The future needs to be invented.

Thus, this proposed work is aimed at the exploratory development of a concept of an adjustable flow of oil pump which can be differentiating in respect of current concepts, looking for solutions that promote greater efficiency, energy, weight reduction, a decrease in the number of moving parts, this maintaining the necessary reliability.

1.3 Document Structure

This dissertation is divided in four chapters in addition to this introductory one.

In Chapter 2 is done the revision of the state of the art where the main hydraulic pumps designs are analysed in its operating principle, main characteristics and applications. Is then analysed specifically the application of variable displacement oil pumps to the automotive market.

Chapter 3 corresponds to the process for product development. Study of current solutions and application with benchmarking to main applications similar to the one to develop and analysis of existent variable displacement oil pumps systems, based in patents. Are then identified the main characteristics to obtain according to the market/client needs and respective specification for the pump and analysis of these ones (QFD). After this process are presented the several concepts developed, the analysis/evaluation of these ones and decision on the one to develop in the dissertation.

Chapter 4 presents the mechanical design of the concept developed with the explanation of the initial analysis for concept specifications/dimensions. Are then reviewed the main components designed and explained the functionality of each one. Finally is analysed the prototype.

Chapter 5 corresponds to the final conclusion on the work developed and definition of future work.

Attachments A to D present some documentation relative to the product development process, the BoM (bill of Materials) and the assembly and definition drawings.

Chapter 2

State-of-the-art Review

2.1 Hydraulic Pumps

Hydraulic pumps are machines/equipments driven by an appropriate motor, their use is directed to fluid propulsion, from a low state of static pressure to a superior pressure state. These can be classified according to the type of operating principle or operating regime.

Concerning the operating principle these can be classified as volumetric or centrifugal.

Volumetric Pumps: The volume variations ensure the displacement of the fluid. These can be:

- Alternate
- Rotatory

Centrifugal Pumps: Ensure the transformation of velocity (rotational) into pressure.

According to the operating regime these can be divided into two groups:

- Fix displacement
- Variable displacement

Regarding the fix displacement, the dispensed volume per rotation is constant and proportional to the rotational speed applied to the shaft. The variable displacement ones possesses a system that allows the control of the dispensed volume per rotation, this system can be mechanical or as more recently used, electro-hydraulic.

There's a larger scale of hydraulic pumps, however regarding the handle of fluids, minimum leakage must be ensured, so, in order to respond to that need, the more commonly used pumps are the rotatory positive displacement pumps. The positive displacement designation means that at each rotation of the pump shaft, a certain volume is displaced to the hydraulic apparatus. The volume varies accordingly to the volumetric efficiency, mechanism dimensions, fluid in motion and operating pressure. Is common to use a safety valve to avoid damage in the mechanism in case of too high pressure.

A variable displacement pump operates by retaining a certain fix volume in the admission/suction zone (inlet port) and then forcing (increasing the pressure) of the same through the expelling zone (outlet port).

The more commons variable displacement pumps are the ones which the mechanism consists in piston/plungers, gears and vanes. For specific applications it is possible to resort to a worm screw or even to systems which incorporates several of the previously referred ones. The displacement of a pump is defined as function of the swept volume in one rotation, this way the flow rate varies with a directly proportional to the system speed. In Table 1 are presented general values for the more common designs of positive displacement pumps at typical operating conditions with mineral oil. All the referred designs are better explained ahead. Notice that the values cannot be seen as precise values, serving only as a general reference in order to foresee the potentiality of each design.(1)

Table 1. Reference values for constant operating conditions and mineral oil. [The Hydraulic Handbook] (1)

Pump Mechanism	Pressure Range [MPa]	Flow Range [l/min]	Speed Range [rpm]	Efficiency (%)
Pistons (inline)	105	High	800	Volumetric 99 Global 90
Pistons (Radial)	35	High	4000	Volumetric 95 Global 90
Pistons (Axial)	35	Medium	3600	Volumetric 95 Global 90
Pistons (bent-axis)	25	Medium	3600	Volumetric 95 Global 90
Vanes	32	455	2500	Volumetric 94 Global 90
Internal Gears	30	400	6000	Global 60
External Gears	25	Medium	360	Global 60

The next subchapters present an approach to the several existing rotatory pumps, this knowledge is focal for the work to be developed.

2.2 Piston Pumps

Piston pumps are characterized by possess a high volumetric efficiency, besides this they virtually have an infinite volumetric capacity. Comparing to gear pumps also as to vane pumps, the piston ones have a considerably more complex architecture, therefore its use is usually restricted to systems that required high operating pressure. Relatively to this characteristic (pressure) the piston pumps are superior to any other design (1).

The operating principle, apart for the type of piston pump design, its base in the fact of fluid suction occurs when the piston draw back in the cylinder and compression and expelling when this moves back forward. Figure 1 represents schematically the piston compression/expelling effect.(2)

Some of the most common applications are for mobile and construction equipment, it is also common for these ones to be used for marine auxiliary power and metal forming and stamping tools. There are several designs of piston pumps: In-line, radial, axial and bet-axis are the main ones to refer.

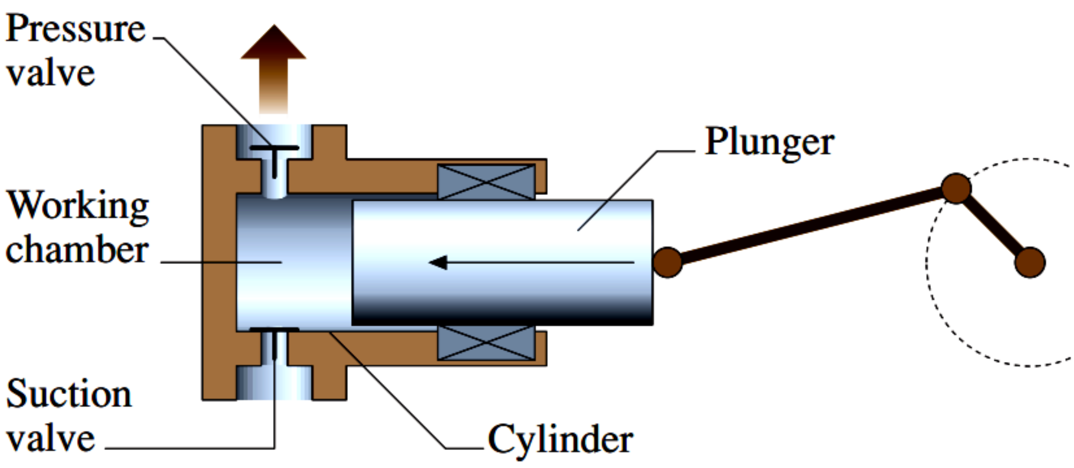


Figure 1. Single Cylinder Piston Pump [Springer Handbook of Mechanical Engineering] (2)

2.2.1 Inline Piston Pumps

Inline (multy-cylindrical) piston pumps are usually robust and designed for applications which require longevity, high pressure and high flow rate. In terms of dimensions these are larger comparing with most of the design. These are the pumps with higher pressure delivery capacity and are also characterized by volumetric efficiency values over 95% even for high flow rate values, the global efficiency is near to 95%. For this type of design the maximum speed is usually restricted to 800[rpm], although there is the referred as ‘high speed pumps’ which can operate at speeds in the order of 1500[rpm].

The mono or single cylindrical piston pumps are simpler in terms of design and intended to operate automatically, this is possible recurring to the differential between inlet and outlet pressure. These are also more commonly used in systems where the working fluid is of a non-lubricant nature.(1)

2.2.2 Radial Piston Pumps

In this type of design, the pistons are disposed radially (perpendicular to the pump shaft axis), Figure 2, is more compact than the inline ones, although possessing a very similar performance, with volumetric efficiency in the order of 95% and global near to 90%, these are much more indicated to variable displacement applications. The limitations of this design in terms of pressure and speed is of 53 [MPa] and 4000 [rpm].(1)

Within this design, two different types can be specified:

- Radial Design: Fixed cylinders block position and rotational swash plate (or wobble-plate).
- Rotational Design: Fixed swash plate and rotational cylinders block.

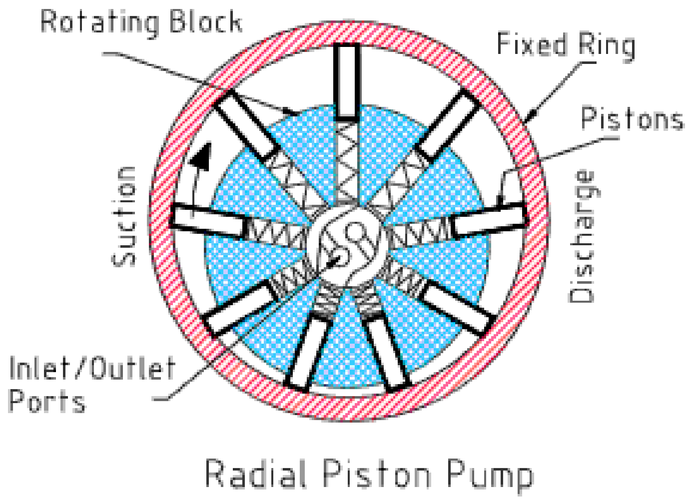


Figure 2. Radial Piston Pump (3)

In the case of the rotational design, the whole assembly rotates around a plate that is static. The pressure control is carried out by the cylinders mass, this fact limits the same to a value in the order of 35 [MPa], also the rotational speed is limited by the total weight of the cylinder block, the maximum reference value is 4000 [rpm]. This configuration can easily be modified in order to ensure variable displacement control, this is achieved with the variation of the plate eccentricity. This design is recurrently used in the in the hydraulic systems applied in maritime and aerial applications.

Opposite to the rotational design, the radial design is characterised by the fixed cylinders block and rotational plate, Figure 2. The drawback movement is carried out by spring or suction effect. The system allow higher rotational speed and pressure when compared with the rotational one, it also allows variable displacement through the variation of the plate axis position.(2)

This rotational design is mainly used in industry for machine tool as presses and injection moulding operations, it is also current to be used for the automotive area.

2.2.3 Axial Piston Pumps

In this layout the cylinders are disposed axially (parallel to the pump axis), these are supported in a central plate which is designed a certain angle (notice that the plate axis is parallel to the cylinders block axis), Figure 3.

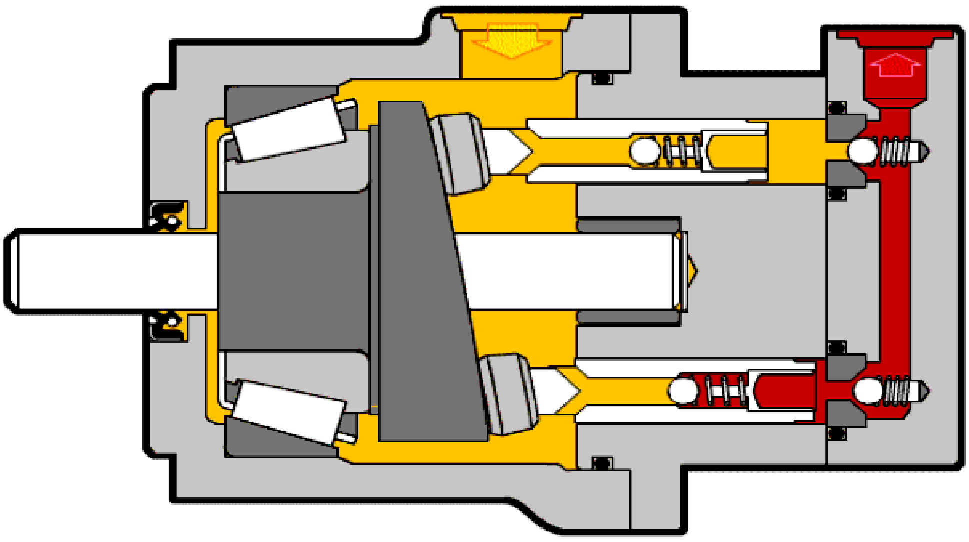


Figure 3. Axial Piston Pump [IHS Global Specs] (4)

The pressure for this type of design is usually limited to a value in the order of 35 [MPa], the rotation speed can go up to 3600 [rpm].

Relatively to the efficiency of these, the reference values are similar to the ones referred for the radial piston pumps (95% volumetric and 90 global). As in the radial layout, there are two possible configurations (the difference is the same referred for the radial design), again, both can be adapted for variable displacement delivery. In this design the more common layout is the one with wobble-plate, this is developed usually for applications that require fixed displacement. Another possible variance in the referred design is the possibility of a second plate where the cylinders are supported, this second plated is guided by the first one through rolling bearings.

The layout with the fixed plate and rotational cylinders block is easier to adapt for variable displacement, on the other hand this systems can become more complex in mechanical construction, this due to need to ensure constant contact between pistons and plate, this is also one of the fact that limits the maximum allowable pressure.

One of the main advances that have been achieved in this type of designs in the possibility of increase the cylinders/pistons dimension keeping the mass value through improvements in the material used. These are currently used for airspace and aerospace application.

2.2.4 Bent-axis Piston Pumps

The bent-axis design is one of the more common when variable displacement is required. This system implies that the plate is assembled with a fixed angle relatively to the pump cylinders block axis (as opposed to the axial and radial design ones). Figure 4.

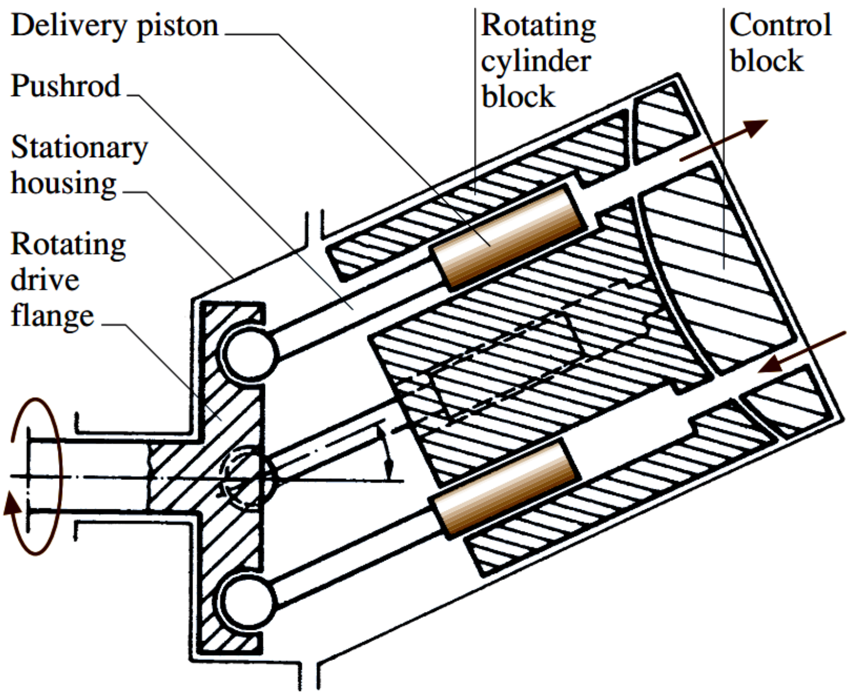


Figure 4. Bent-Axis Piston Pump [Springer handbook of Mechanical Engineering] (2)

The pistons are connected to the plate through a spherical joint, this way it is possible to ensure the mechanical connections and keep the assembly angle. The operating principle is the same as in any other piston pump.

This layout can also be configured to deliver a variable displacement, that's achieved by changing the angle between angle and pistons cylinder axis.

This design is usually limited at values of 25 [MPa] in pressure and 3600 [rpm] in rotational speed and its common to be used for vehicles applications.(2)

2.3 Vane Pumps

Vane pumps are particularly suitable for medium operating ranges (in terms of pressure and speed). These have as main advantage, especially comparing with conventional gear pumps, the fact of being hydraulically balanced, this way the loads that the mechanical system is subjected are considerably diminished. The versatility also as the range of rational speed as lead to become a viable alternative to gear pumps. Previously his major application was confined to systems that required medium pressure values (for example machine tools). Figure 5.(2)

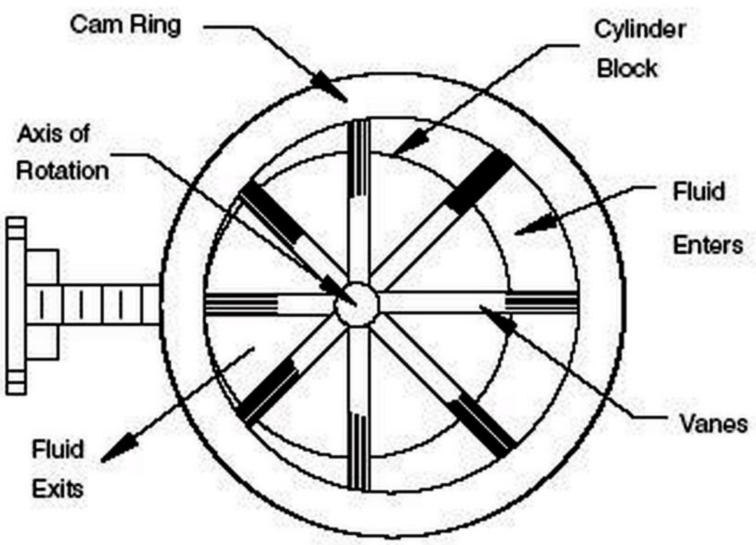


Figure 5. Vane Pump scheme [Hydraulic Pumps]

Currently this design covers a large range of possible rotational speeds, as reference it can be referred a minimum speed of 200 [rpm] and a maximum of 2500 [rpm] (this for the generality of the applications, it can go higher for specific designs), the pressure values can go up to 32 [MPa]. However when considering variable displacement vane pumps, the reference value for maximum pressure if of 7 [MPa].

The vane pumps referred as single-cell are hydraulically not balanced, opposed to the more common multi-cell design. The fact of being not balanced (the singe-cell ones) leads to a low volumetric efficiency, this is caused by the difficulty to control internal leakages, even so this type of layout presents the advantage of being easier to adapt for variable displacement delivery. The variable displacement is achieved by adjusting the eccentricity of the external/cam ring relatively to the axis of rotation (that carries the vanes), this way the volume swept for each rotation between the inlet and outlet zones varies. The solution to contradict the non-balanced issue is to use multi-cell layouts, this means that the two or more vanes share the same rotor, thereby, increasing the number of cells/pockets (gap between two vanes), this design is more complex. An additional solution to avoid the non-balance is to use two inlet and two outlet ports, these being disposed in opposite sides of the rotor.

The more common design is the multi-cell fixed displacement one, Figure 5, the main reasons is the fact of possessing a higher volumetric efficiency and the capacity to deal with higher pressure values (comparing with single vane ones).

The operating principle of these consists in the transport of fluid in cells (between each two vanes) from a low pressure state to a higher pressure. The fluid in each cell is compressed with the movement of the cells from the inlet to the outlet zone, the same amount of fluid sucked in the inlet into one cell is dragged to a smaller cell in the outlet, this way the pressure increases and the fluid is expelled. This is achieved due to the eccentricity of the rotor and the fact of the vane being permanently in contact with the cam ring wall. Instead of an eccentric rotor it is possible to use an elliptic cam ring which creates the same variable cell size effect with the vanes rotation.(1)

Some of the more common applications for these are die casting and moulding machines as well as road construction machinery.

2.3.1 Fixed Displacement Vane Pumps

This layout finds in the automotive industrial area its major application. The operating principle consists in a central rotor which drags several vanes, these vanes sweep the volume contained in the cam ring, Figure 6. Due to this design, different flow rate ranges imply different vane/Cam ring sizes. (1)

Figure 6. Internal block of a fixed displacement vane pump. [HOF Hydraulic] (5)

One of the major issues in this design is the wear suffered by the vanes, this occurs due to the need of ensuring that these ones are constantly in contact with the cam ring walls. A solution to diminish this wear is to use double-lip vanes (for each vane there are two surfaces in contact with the cam ring wall).

There are two main solutions to ensure the constant contact between vanes and cam ring, either the vanes are forced out of the rotor by a spring, or, the layout is designed in order to recirculate fluid from the outlet zone into the base of each vane in the rotor, with this, the pressure in the base of the vane combined with the centrifugal force caused by the rotation ensure the vanes positions against the cam ring. This solution (with recirculated fluid) leads to a smaller wear of the components and also enables the system to work at lower pressure and rotational speed values.(1)

2.3.2 Variable Displacement Vane Pumps

Opposite to what occurs in the fixed displacement vane design, the variable displacement ones are not balanced, this is due to the fact of the radial movement of the cam ring relatively to the rotor, the variable displacement is achieved by this movement. The motion of the ring is usually controlled by a spring that is pre-forced in the external surface of the cam ring in one end and is loaded by fluid at outlet pressure at the other, this differential leads to the cam ring movement and consequentially a variable displacement per rotation. The cam ring movement is limited between the most eccentric position and a near concentric one. Figure 7. (1)

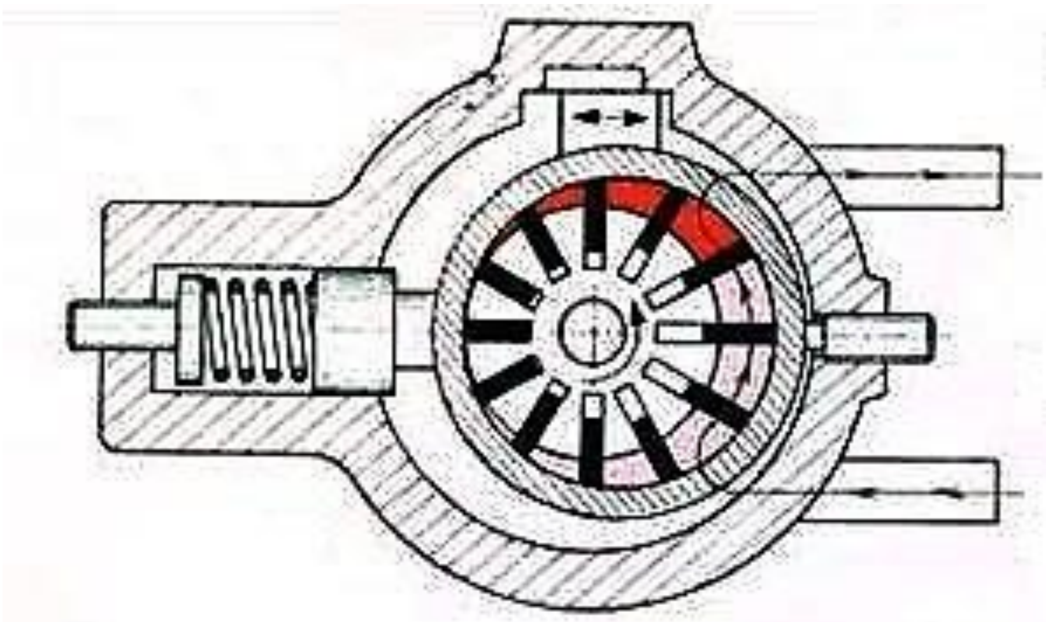


Figure 7. Variable Displacement Vane Pump [Santec Group] (6)

The fact of the outlet pressure controlling the spring and cam position means that is possible not to use a safety discharge valve (not in every application). When pressure rises, the cam ring is forced to a more concentric position which leads to an outlet pressure reduction, this way, theoretically, the situation of over pressure never occurs.

2.4 Gear Pumps

The Gear pumps are one of the most common designs, these can have mainly two different mechanism possibilities, external (two external gears) and internal (one external and one internal gear). The gears teeth design can vary considerably from the usual (involute shape), this is needed in order to ensure maximum tightness (no leakage) and efficiency.

The operating principle consists in the drag of fluid into the gap between each two teeth and the pump housing. The oil is dragged and forced to the outlet due to the gears rotation. Figure 8 and Figure 9. (1, 7)

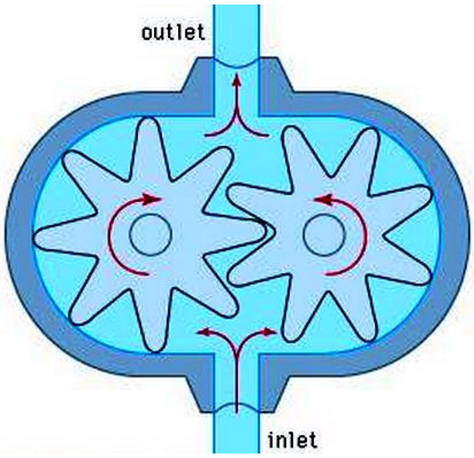


Figure 8. External Gear Pump scheme [Encyclopaedia Britannica] (7)

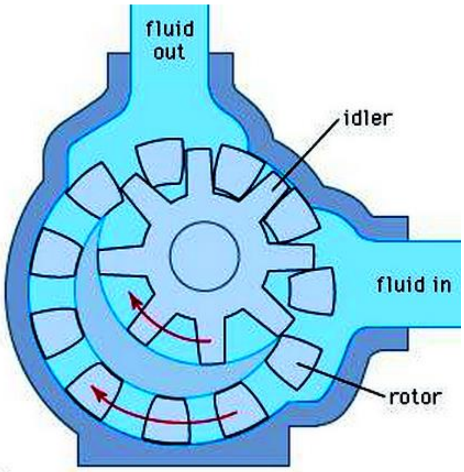


Figure 9. Internal Gear Pump scheme [Encyclopaedia Britannica] (7)

In the external gears layout the fluid circulation occurs between the teeth and the pump housing. In the case of more conventional internal gear pumps the fluid is moved between the pair of gears (internal and external) and a mechanical filler/crescent.

2.4.1 External Gear Pumps

This design is the one with a larger range of applications, comparing with vane pumps, these are capable to operate with higher rotational speed and similar pressure. The pressure can go up to 30 [Mpa] (again, just a reference value for general application), the speed can easily go to 6000 [rpm]. The efficiency of this layout is in the order 60% in global, it can go up to 95% in volumetric efficiency for precision gears. The two gears have usually the same diameter and teeth design, the inlet and outlet ports are disposed in opposite sides of the meshing zone, Figure 8. One of the gears is driven from an external system/engine while the other is driven by the first one, both are support by low friction bearings. Fixed displacement pump, Figure 10. (1, 8)

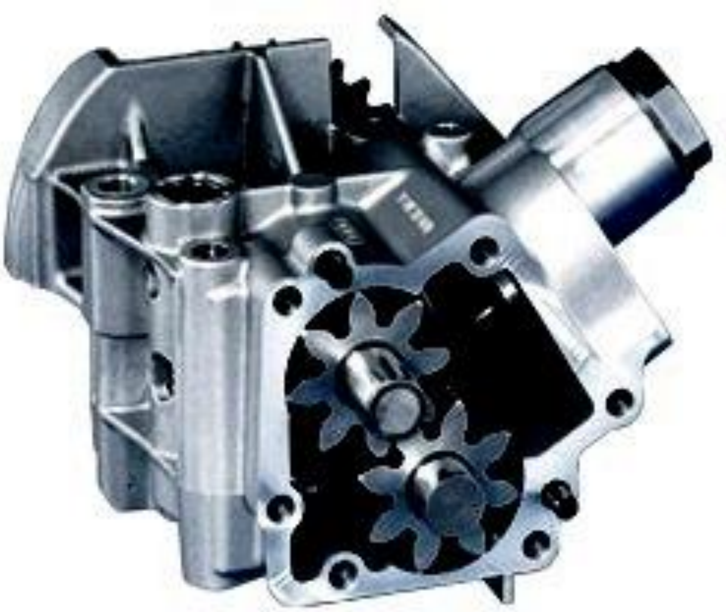


Figure 10. External Gear Pump [SHW fixed displacement pumps] (8)

The operating principle consists in the fact of the gear rotation lead to reduction of fluid volume in the inlet zone, this way is created a suction effect that due to the gears rotational direction forces the fluid to the outlet port. The fluid is dragged on the outer side of the gears, being restricted between the gears and pump housing (pockets).

In order to maximize the volumetric efficiency, the leakages between pockets must be as reduced as possible, this leads to non-balanced loads applied to the bearings which reduce the mechanical efficiency of the system. The solution is usually to find a balance solution to achieve the maximum global efficiency possible, however is common to compromise mechanical efficiency in order to achieve a higher volumetric efficiency once this one is more preponderant in the global behaviour. This fact is the reason for the mechanical efficiency in this design being typically lower than the remaining ones. Usually side plates are used to avoid leakage without large losses in mechanical efficiency. The plates are disposed in the lateral surfaces of the gears, these are pressurized against the gears walls using the pump fluid itself, this way the applied force is proportional the interior pump pressure.

The balance can be optimized adding external recirculation to the gears, however this flow must be with a low pressure value, this can be achieved with a certain clearance in the engagement, also, to allow this recirculation the solution is to add grooves in the lateral surfaces of the gears, usually this is only adopted when high pressure is required.

One of the major advantages of the external gear pumps is the capability of delivery low flow rates per rotation, this is ideal for applications with small dimension that require high pressure values. These are usually used for lubrication in machine tools, fluid power transfer units and automotive applications.

As referred the gears are usually non-involute, this is needed to ensure higher displacement and efficiency values. The changes to the gear shape must be carefully analysed to avoid leakage between teeth. An usual choice is to use helical teeth shape, these allow a less noisier system, with higher delivery per rotation without losing efficiency.(1, 2)

2.4.2 Internal Gear Pumps

Conventional Internal Gear Pumps

The basic design of this pumps consists in a gear (internal gear) positioned in the outer ring, this engages with an external gear. The gears are designed in order to ensure that each teeth of the internal gear is engaged with the internal gear. The pockets that drag the fluid from the inlet zone to the outlet zone are created between both internal and external gear and a solid shape (filler) that is disposed in the non-engagement area. Figure 11.

Some of the advantages that this system brings comparatively to the external gears one are the lower loads applied in the mechanism, the fact of be more silent and also the capacity to easily do adapt do multi-stage (two rotors, usually assembled in the same axis).

In the majority of the applications, the external gear drives the internal gear (external ring gear).Some of the more common applications for these are non-mobile hydraulics such as machines for plastic and machine tools and vehicles that work in an enclosed space. The reference values for maximum admissible pressure and rotational speed are 25 [MPa] and 3600 [rpm]. (1, 2)

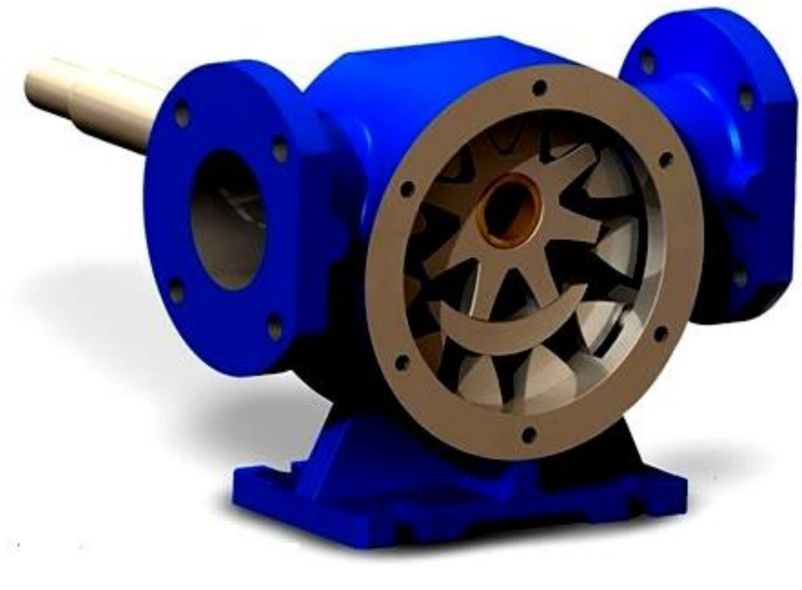


Figure 11. Conventional Internal Gear Pump [King Pompa - M Series] (9)

Gerotor Internal Gear Pumps

The gerotor gear pumps have the same operating principle of the conventional ones, however these do not have filler.

One of the facts that distinguishes this type of pumps is the extremely simple design and consequently low cost, for this reason these are many times used as secondary pump in an hydraulic system, as an example de pre-pressure of fluid prior to a piston pump (supercharge effect). These have a special design that allows the drag of fluid without the filler, this is possible the contact between surfaces in the non-engagement zone.

Gerotor pumps are limited to values near 14 [Mpa] and 1000 [rpm] in terms of pressure and speed.

Figure 12. Gerotor Internal Gear Pump [Feuling 7060] (10)

2.5 Application of variable displacement oil pumps to internal combustion engines

As referred the oil pumps used for internal combustion engines in the automotive area did not suffer significant changes for a long period, mainly due to his robustness, simplicity and the importance of these for the engine integrity. However in the last few years, due to needs in the area of efficiency and CO_2 emissions reduction, being the oil pump an essential element in the engine/car dynamic and also a relevant power consumer part, has been target of changes to fulfil the exigencies.

The usual (fixed displacement) oil pumps used in the automotive area are mainly external gear and gerotor designs, however when the variable displacement is required for this applications the most used is the vane pump design. This is due to the simplicity of the design (in variable displacement) of the vane pumps such as the reliability and the flexibility to adapt to different working requirements (rotation speed and pressure). With the increasing utilization of this variable displacement design in the area: several issues related with the utilization of these have also been analysed. Some of the more usual problems are directly related with the mechanical design of these pumps are the high wear suffered by the vanes and internal wall of the pump.

The use of the variable displacement pump in an internal combustion engine system influences several aspects of the engine/car, binary and power consumption, pressure and flow delivered, gas emissions and fuel consumption among other are some of the most important.(11, 12)

2.5.1 System/Engine Requirements

The oil pumps needs to ensure the delivery of fluid within a required range of flow and pressure in order to fulfil several requirements (Figure 12):

- Lubrication and cooling of the engine and auxiliary systems as camshaft, crankshaft, connecting rods.

- Counteract centrifugal forces of the crankshaft.
- Drive auxiliary systems (with oil pressure activation)

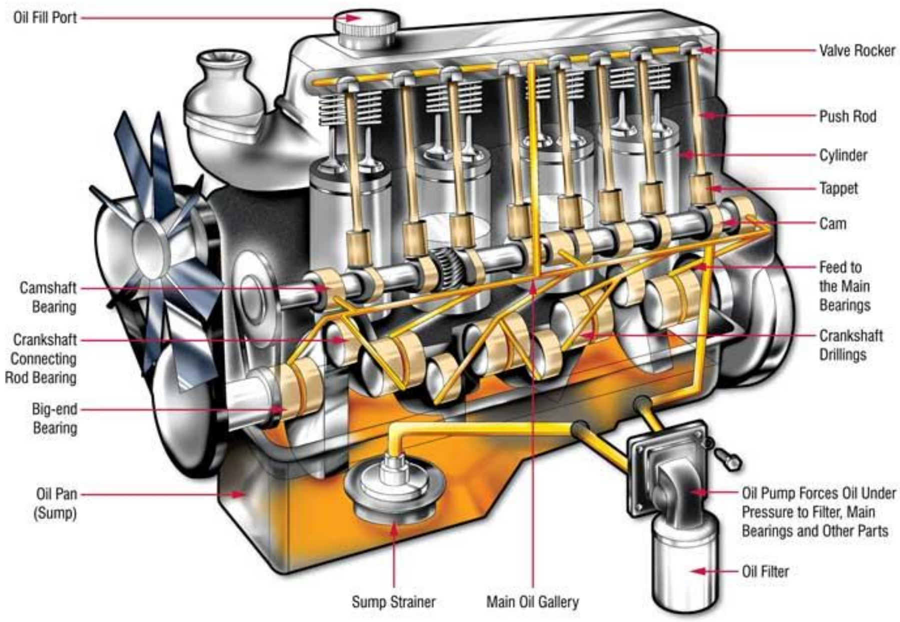


Figure 13. Oil delivery in Internal Combustion Engine – scheme. (12)

2.5.2 Pump drive system

One of the factors that most influence the whole dynamic of the lubrication system is the way as the oil pump is driven (application of torque to the pump shaft), the different options possible have implication in the pump power consumption and general efficiency. Typically, in traditional designs, by reliability reasons, the pump is direct driven by the crankshaft, however in order to improve the energy consumption of the pump arises the need to use the referred as off-axis systems, with driving chains or belts.

The graphic of Figure 14 represents the Mechanical Efficiency (Meff) variation for an oil pump in the same working range (flow and range) with different driving applications. In the vertical axis it is represented the Meff (Mechanical Efficiency [%]) and in the horizontal one the engine rotational speed (Engine Speed [rpm]). The graphic is based in a study carried out in a 1.0L engine.

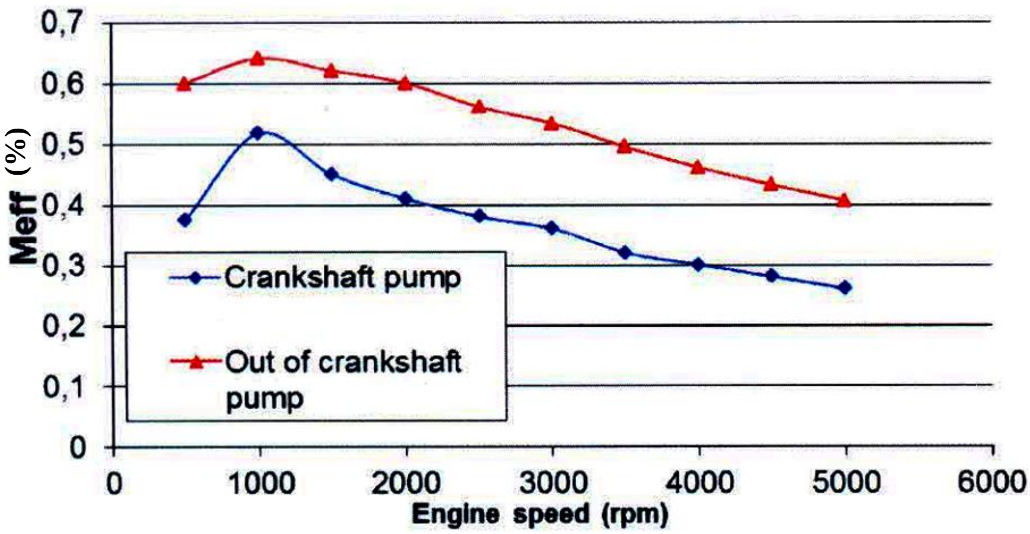


Figure 14. Mechanical Efficiency (Meff) of oil pumps with different driving designs for the same flow delivery. [Strategies for Energy Savings with use of Constant and Variable Displacement Oil Pump Systems](11)

For the specific case in analysis there is a near to 20% difference in mechanical efficiency for all the rotational speed range, although being a specific case, this clearly shows that there is a substantial improvement when using off-axis driven systems, therefore this is one of the most important factors to take into account.(11)

2.5.3 Volumetric Efficiency

In the global behaviour of the pump one of the most relevant factors is the volumetric efficiency. The global efficiency is a sum of mechanical and volumetric factors, being the volumetric the one with more influence.

This way, as referred, in the variable displacement designs, the fact of fluid recirculation due to excess of flow not being needed improves significantly the volumetric efficiency when comparing with fixed displacement systems. This is even more important and differentiating when the rotational speed is high since it is in this speed range that the overflow is more relevant. The graphic of Figure 15 shows the difference in volumetric efficiency for two different pumps (variable displacement and fixed displacement) for the same system needs and speed range. In the vertical axis is represented the volumetric efficiency [%] while in the horizontal is the rotational speed (Engine Speed [rpm]). (11, 13)

The graphic is based in a study carried out in a 1.0L engine

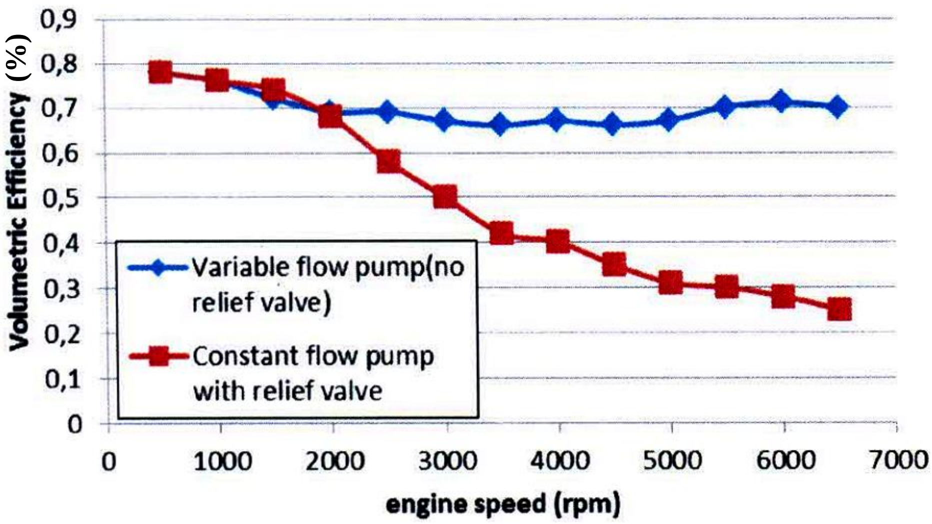


Figure 15. Volumetric Efficiency of variable and fixed displacement pumps for the same speed range. [Strategies for Energy Savings with use of Constant and Variable Displacement Oil Pump Systems](11)

The graphic clearly shows a discrepancy in volumetric efficiency of the two pumps with the rotational speed increase. In the variable displacement pump the volumetric efficiency decreases approximately 10% with the increase of rotational speed from 1000 [rpm] to 7000[rpm] while in the case of the fixed displacement pump the decrease is of 50%.(11)

2.5.4 Binary/Torque Consumption

Two of the most relevant factors in a vehicle engine are the torque and power delivered, this way is rather important to optimize the use of these ones, meaning, decrease the consumption by auxiliary systems in the car as the oil pump. In terms of binary consumption it is estimated that approximately 2,5% of the total developed by the engine are delivery/used by the oil pump system (considering a regular fixed displacement system).Table 2 presents the data for a specific case with an 1.0L engine at 50% load.(14)

Table 2. Analysis of consumed torque by oil pump (for internal combustion engine with 1[l] at 50% of full load) [Strtegies for Energy Savings with use of constante and variable Oil Pump Systems]

Rotational Speed [rpm]	Engine Torque [N*m]	Oil pump Torque [N*m]	Consumed Torque [%]
1000	63	0.32	0.54
1500	73	0.59	0.81
2000	74	0.89	1.20
2500	78	1.09	1.40
3000	83	1.19	1.43
3500	80	1.28	1.60
4000	83	1.38	1.66
4500	85	1.50	1.76
5000	82	1.59	1.94
5500	78	1.67	2.14
6000	72	1.77	2.46

Although for different operating conditions relatively to the Table **2**, the graphic of Figure 16 represents the different behaviours of several oil pumps (with different designs/operating principle) including fixed and variable displacement. In the vertical axis is represented the absorbed torque (Torque consumed by the pump [Nm]), in the horizontal axis is the pump rotational speed (speed [rpm]). The graphic is based in a study where were used three different pumps, for the case in study the interest is to analyse the 'variable displacement' and the 'fixed displacement' ones, these where submitted to the same conditions and required to deliver the same flow at each step.

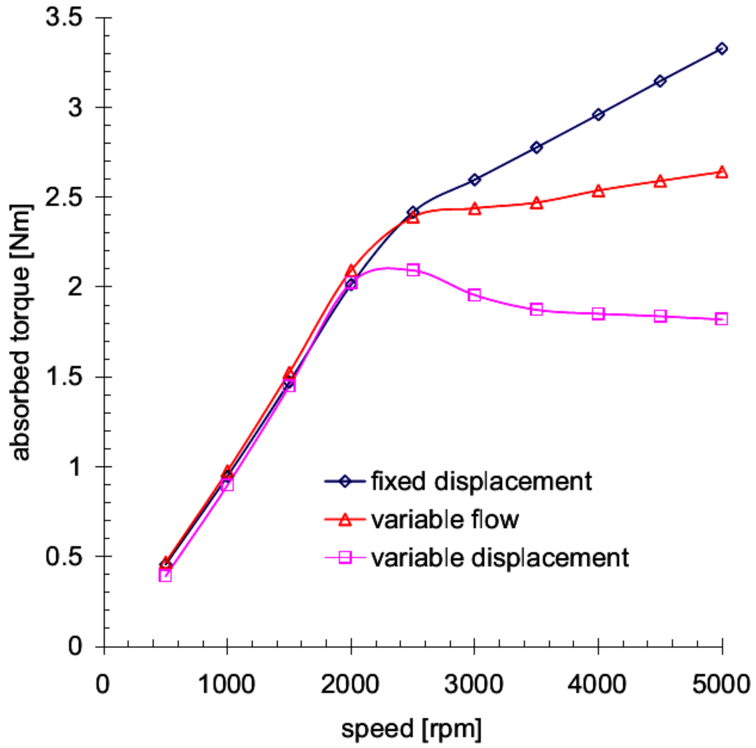


Figure 16. Consumed torque for different oil pumps (different designs) [Displacement vs Flow control in IC Engines lubricating pumps](14)

Analysing the graphic it is possible to observe that the fixed displacement oil pump torque consumption increases in direct proportion to the rotational speed. However, in the variable displacement oil pump the torque consumption is very similar to the fixed displacement until the 2500 [rpm], from that value up the evolution of the torque consumption is the opposite to the fixed displacement once it starts to decrease until a near constant value. This is due to being in higher rotational speeds that the variable displacement non necessity of fluid recirculation in a more differential factor. (14)

2.5.5 Power Consumption

The power consumed by the oil pump is also a relevant factor, this is directly related with the oil pump mechanical efficiency and the system needs in pressure and flow.

Once the fixed displacement pumps are not able to adapt to the system needs, these are dimensioned in order to fulfil the system maximum requirements even at lower rotational speeds (critical conditions for the pump) this leads to over-delivery for higher rotational speeds, and so, an excess of power consumption. In the variable displacement pumps, due to the adaptability to the system needs, the power consumed at each step is the strictly needed.

The graphic of Figure 17 represents qualitatively the excess of power consumption verified for fixed displacement oil pumps contrary to what happens for variable displacement designs. In the vertical axis is represented the power consumed, the horizontal axis presents the rotational speed.(15)

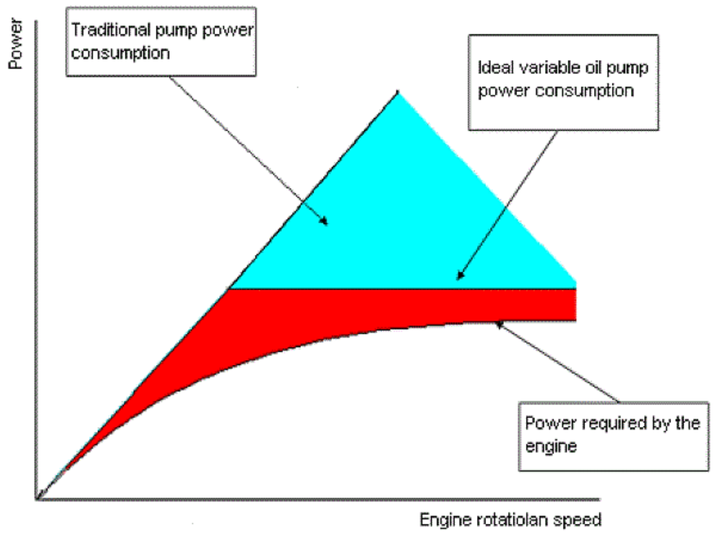


Figure 17. Power consumed by fixed displacement versus variable displacement (vane) oil pump. [Numerical and Experimental Analysis of Experimental Vane Pumps] (15)

Is visible that for the fixed displacement pump, the power consumption is directly proportional to the speed increase which does no goes 'in line' with the engine requirements at each step, for the variable displacement pump the power is near to the required by the engine with the rotational speed increase. The graphic of Figure 18 presents the difference of power consumption of gerotor and vane pumps with different designs each (fixed and variable displacement), is still distinguished the pump driving method (direct or off-axis) for each one. In the vertical axis is represented the power consumed (Crankshaft Power [W]), in the horizontal one the rotational speed (Speed Engine [rpm]).(11)

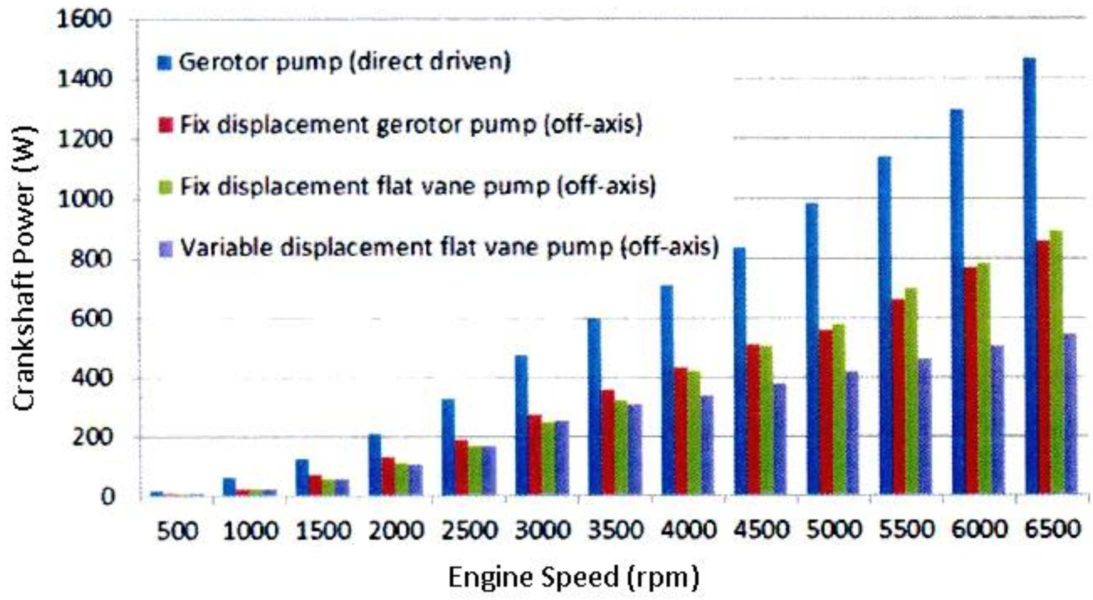


Figure 18. Power consumed by fixed displacement versus variable displacement oil pumps with differentiation between direct driven and off-axis. [Strategies for Energy Savings with use of Constant and Variable Oil Systems] (11)

From the analysis of both graphics is possible to conclude that the power consumption is highly reduced (mainly for higher rotational speeds) when using variable displacement designs.

2.5.6 Pressure and Flow Rate

The pressure and flow rate are the requirements that need to be ensured by pump system, these vary proportionally, this way it is possible to do an analysis of both factors in an integrated manner.

The graphic of Figure 19 represents the pressure and flow rate delivered by a variable displacement vane pump depending on the rotational speed value. In the left vertical axis is represented the pressure [bar] which applies to both pressure lines represented (inlet and outlet), the right vertical axis presents the flow rate value [l/min] which is relative to the flow rate represented line, the horizontal axis presents the rotational speed [rpm] values relative to all the variables.(14)

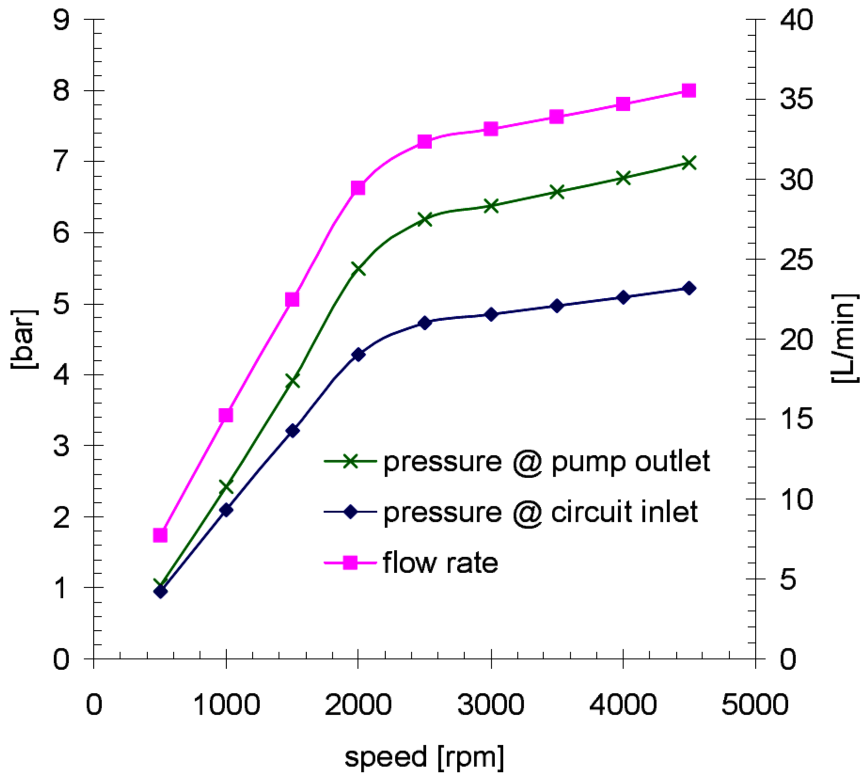


Figure 19. Pressure and flow rate relation to rotational speed for variable displacement vane oil pump. [Displacement vs Flow control in IC Engines Lubricating Pumps] (14)

The graphic of Figure 20 shows the significant over-delivery of fluid by a conventional fixed-displacement pump for higher rotational speeds. The vertical axis represents the flow rate value (required flow rate [L/min]) while the horizontal one the rotational speed (engine speed [rpm]). In the graphic are represented the maximum and minimum flow rate values for the speed range values, between these is the system required flow rate. The specific values represented in the graphic are based in the general requirements for a four cylinder Diesel engine. (14)

The excess of flow rate from the fixed displacement pump is compensated (not delivered to the system) being recirculated on the discharge valve, this represents a waste of energy, in the variable displacement design this compensation is not needed.

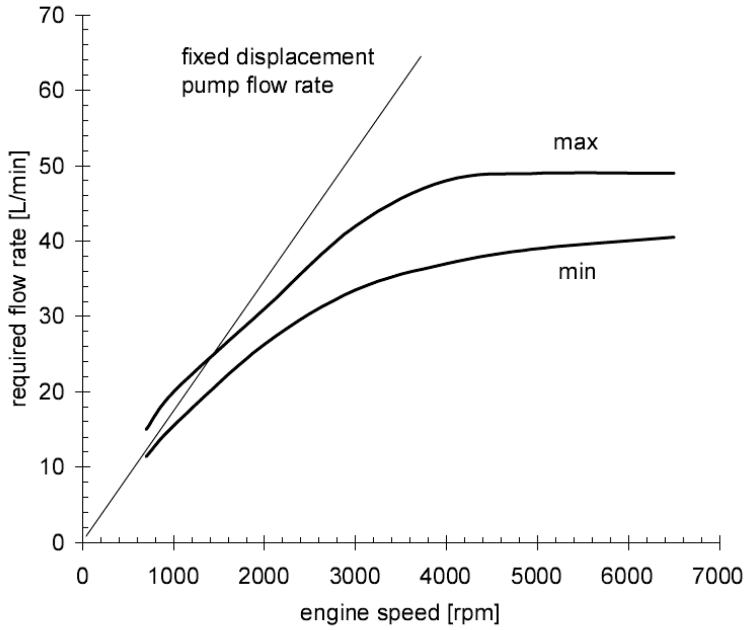


Figure 20. Required versus delivered flow rate for conventional fixed displacement pump. [Displacement vs flow control in IC Engines lubricating pumps] (14)

2.5.7 Fuel Consumption and CO_2 emissions

One of the main motivations for the development of variable displacement oil pumps intended to be used in internal combustion engines is the limitation in CO_2 emissions that have been imposed in the last few years, this value is currently restricted by the Euro 6 norm that defines for passenger cars a value of 170mg/km. The CO_2 emissions are related/dependent of the whole vehicle dynamic, from the engine to all the auxiliary systems.

One example of the contribution that an more efficient oil pump can bring to the CO_2 emissions reduction is the recent engine from Renault (1.6 DCI 130) which according to Renault have a 1% reduction on emissions only due to the use specially designed variable displacement oil pump.

The emissions are directly related with the fuel consumption, the new Audi engines (2.8 FSI and 3.2 FSI) use a variable displacement pump which according to Audi allowed fuel consumption saving in the order of 5%. Studies realized to evaluate the potential in fuel savings with the use of a variable displacement oil pump reveal that the savings are (for general cases) between 1.5 and 2.5%. (14)

The graphic in Figure 21 represent the difference in delivered flow for a fixed displacement and a variable displacement oil pumps and the consequent percentage in fuel savings from the use of the variable displacement system, this for different vehicle speeds [Km/h], in the performed study it was kept the requirements for oil flow and pressure. The green line in the graphic represents the perceptual fuel savings when using the variable displacement oil pump comparatively to the fuel consumption with the fixed displacement one.(11)

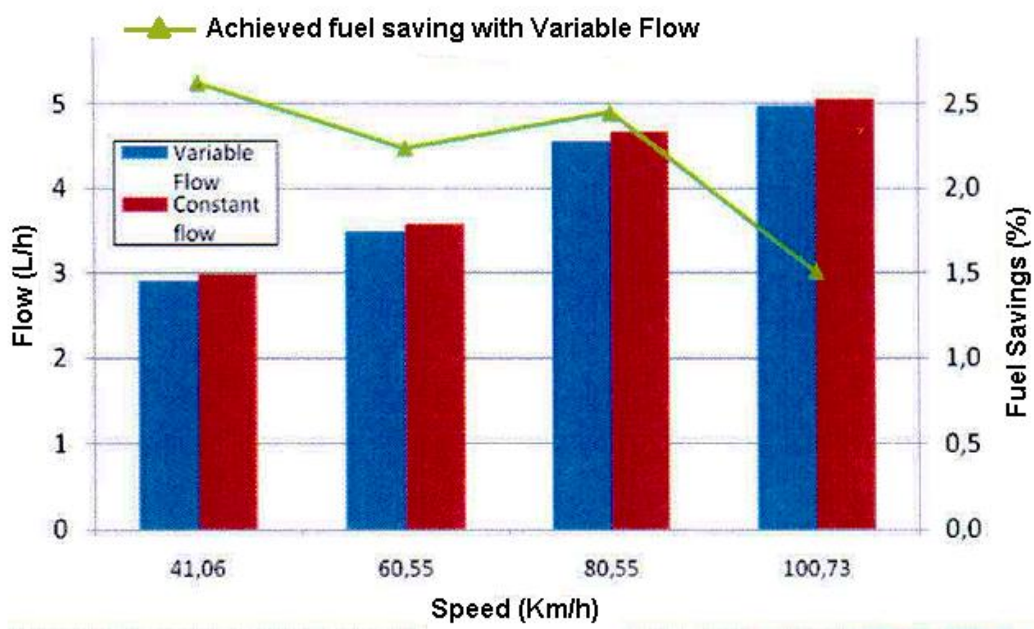


Figure 21. Fuels savings achieved comparing the use of fixed and variable displacement oil pumps. [Strategies for Energy Savings with use of Constant and Variable Oil Pump Systems] (11)

Chapter 3

New Product Development Process

In order to ensure that the product development if focused in the client/market need, a logical and standard process must be followed.

The development/definition process goes through several phases, from the Benchmarking to the industrial design definition:

- Benchmarking.
- Identification/Classification of the Client needs.
- Identification/classification of the product requirements (Mudge Diagram / Kano Diagram).
- Identification/classification of product specification (QFD Matrix).
- Splitting the product development issues in sub-parts.
- Concepts generation (architectures)
- Classification of concepts/ concept selection
- Industrial design

3.1 Benchmarking Analysis

The benchmarking analysis that was carried out was not focused in technical specification since such information is restricted to secrecy by the entities that detain the products. This way the analysis is on the effect/advantages brought by the use of the variable displacement pumps in the automotive market.

Just as a general overview are referred the main companies focused in the variable displacement pumps market also as the general characteristics for the global majority of these products.

Variable displacement oil pumps companies:

- SHW (Variable displacement vane and gear pumps for the automotive market)
- Pierburg KSPG (Variable displacement multi-stage pumps for automotive market)
- Scherzinger
- TCG Unitech
- Concentric
- SLW Automotive
- PMG

General Characteristics:

- Weight (median) = 2.3 Kg
- Material (aluminium body with internal elements in steel)
- Achieved CO_2 emissions reductions = 1%
- Achieved fuel consumption reductions = 1.5%
- Price = 250€

From the variable displacement oil pumps used for internal combustion engines, the vane ones stand out. Several automotive brands have been used variable displacement oil pumps and reported the evident advantages of the same ones:

- Chevrolet® Ecotec:

Several from the Ecotec range of engines produced by GM (General Motors) to be incorporated in Chevrolet vehicles resort to variable displacement oil pump systems.

The Figure 22 represents a variable displacement oil pump used in the Ecotec 1.4L Turbocharged engine. This consists in a vane pump which varies the eccentricity of the rotor axis by moving the outer ring which rotates in a fix point, the angle between the two positions (near concentric and eccentric) is represented in the Figure 22.

Also the V8 - 6.2L (Chevrolet Corvette) and the Ecotec3 (4.3 – V6; 5.3 – V8; 6.2 – V8) recur to the variable displacement technology.(16)

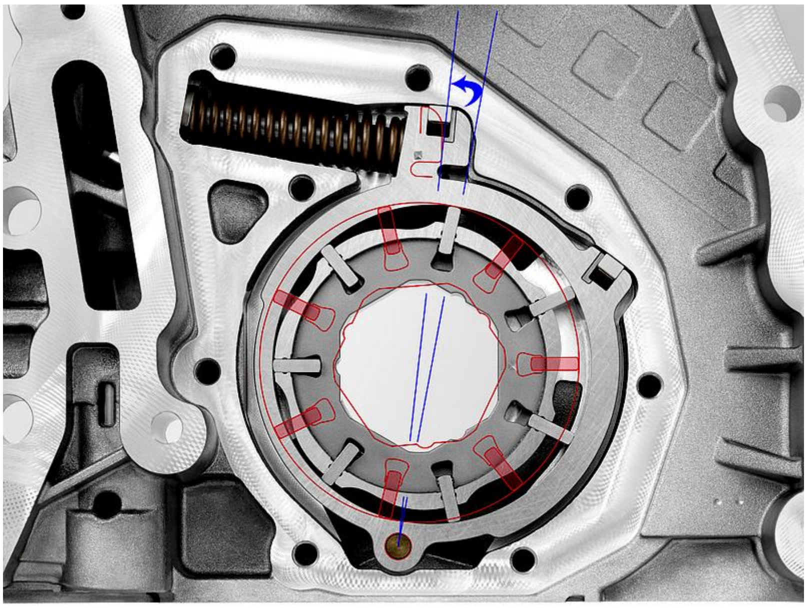


Figure 22. Variable displacement oil pump - GM Ecotec. [Variable Displacement Oil Pump in Chevrolet Cruze Aids Fuel Conservation] (16)

Quoting Mile Katerberg (assistant chief for the Ecotec 1.4l):

"By lowering the volume of oil we reduce the amount of energy, or torque, required to pump the oil, without taking necessary lubrication away from the engine. Reducing the torque demand reduces fuel consumption. It is a simple, durable, maintenance free design that we have used in our transmissions for years and more recently in our hybrid vehicles"

- Peugeot®:

A new segment of engines from Peugeot (PSA – Peugeot – Citroën) intended to incorporate the Peugeot 208 B-Segment' also recurs to this technology. The available information is not specific but this technology is to be incorporated in both diesel (e-HDI) and petrol (3 cylinders) engines (17):

- 1.0l VTi
- 1.2l VTi
- 1.4l VTi

- 1.6l VTi
- 1.6l THP
- 1.4l e-HDi PEF
- 1.6l e-HDi PEF

- Mercedes®:

The new engines that incorporate the A Series also recur to the variable displacement pumps technology. The A180 CDI and A220 CDI present a fuel consumption reduction in the order of 26% relatively to the predecessors, a percentage of this if, according to Mercedes, due to the variable displacement pump which allows a optimization of the oil flow.(18)

- Ford®:

Some models from the new Fiesta and Focus from Ford have three cylinders with 1l engine (1.0 EcoBoost). These as been announced by Ford as the most economic engine from their range of products. Among the technologies that incorporate this engines is a variable displacement oil pump which according to Ford is computer-controlled, meaning that recurs to an electro-valve controlling the pump behaviour.(19)

- Hyundai®:

The new 3.8 GDI (Gasoline Direct Injection) engine from Hyundai achieves 7% decrease in fuel consumption relatively to his predecessor. This engine recurs to an variable displacement oil pump that, according to Hyundai, allows a better behaviour in oil pressure delivery and parts lubrication.(20)

- Renault®:

The 1.5 DCI 130 from Renault is focused in technologies that allow a better efficiency relatively to his predecessor, among these technologies a variable displacement oil pump. According to Renault, this technology (variable displacement pump, Figure 23) contributes alone with 1% in fuel consumption decrease, this in a total of 20% achieve for the engine.(21)

Figure 23 represents schematically the pump operating principle.

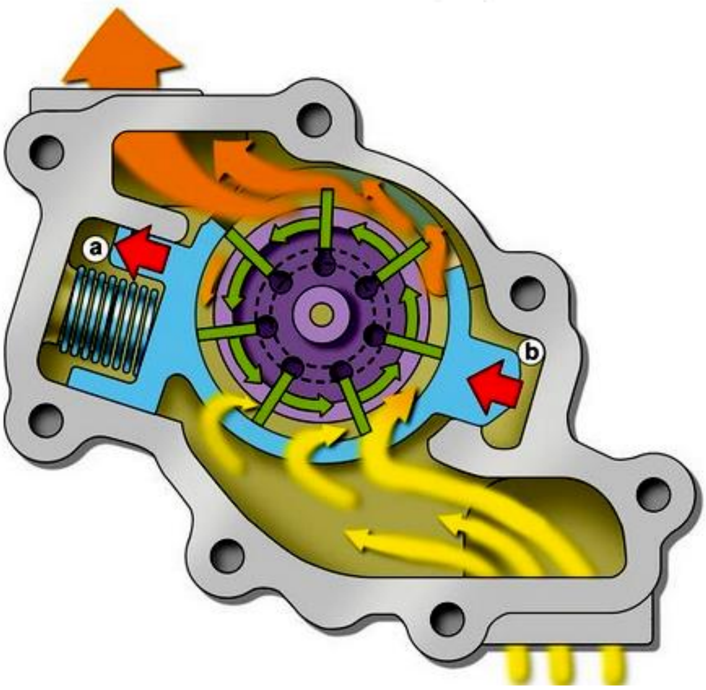


Figure 23. Variable Displacement oil pump - Renault. [Car Advice - Renault new Energy dCi 130 Diesel engine derived from F1 experience](22)

3.2 Variable displacement pumps - Patents analysis

3.2.1 Variable Displacement Oil Pump - US2012/0045355 A1

Patent References (23):

- Patent Application Publication
- Name: VARIABLE DISPLACEMENT OIL PUMP
- Inventor: Paul Morton
- Pub. No.: US2012/0045355 A1

- Pub- Date: Feb. 23, 2012

The concept assumes the operating principle of a conventional vane pump, the displacement variation system is based in the movement of the outer ring, meaning, the eccentricity variation of this relatively to the rotor axis, this way the ratio between the inlet and expelled volume for one single rotation is modified. The differentiating factor in the concept is the method to control the eccentricity/outer ring position.

The rotor position is fixed, the only movement is the rotation that drives the vanes along the pump,this rotation allows the transportation of fluid from the inlet to the outlet port. The external ring movement is restricted to a certain angular variation around a fix point.

In Figure 24 it is possible to identify the referred mechanism elements. The outer ring (3) position varies by rotating around the fixed point (6), this is due to the load applied by the spring (8) balanced with the fluid pressure applied in the outer ring external wall at the gap/channel (7). This movement defines de ring (3) eccentricity relatively to the rotor (2) which is fixed with only rotational movement allowed.

The rotor (2) is driven by the shaft (1), this transport the vanes (5) which due to the centrifugal force and oil pressure in the channels (4) forces the vanes against the outer ring internal wall. In the Figure 24 the pump is represent in the most eccentric position allowed corresponds to maximum displacement per rotation (considering a constant rotational speed). The descending movement of the spring (8) caused by an oil pressure (oil at outlet pressure) load leads to the eccentricity reduction (nearest to concentric), this leads to a displacement value decrease, the less eccentric position is represent in the figure in dashed line.

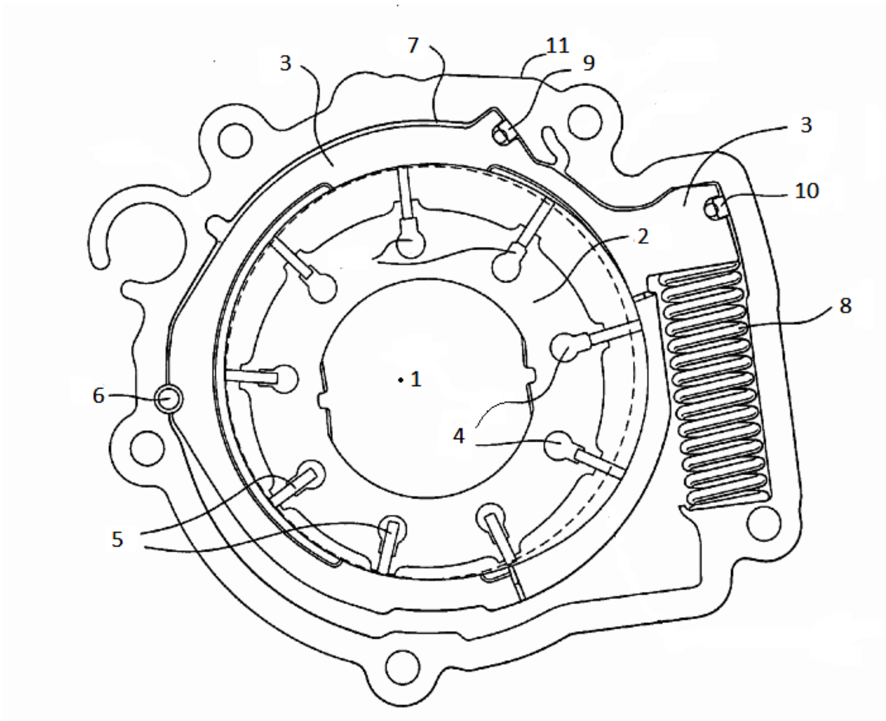


Figure 24. Scheme representative of the concept with eccentricity variation with outer ring movement around fixed point. ***(23)***

Figure 25 schematically represents the different volume dragged at two outer ring positions. In the indicated items in the figure [A] and [B] is possible to see the dragged volume for maximum and minimum displacement positions respectively.

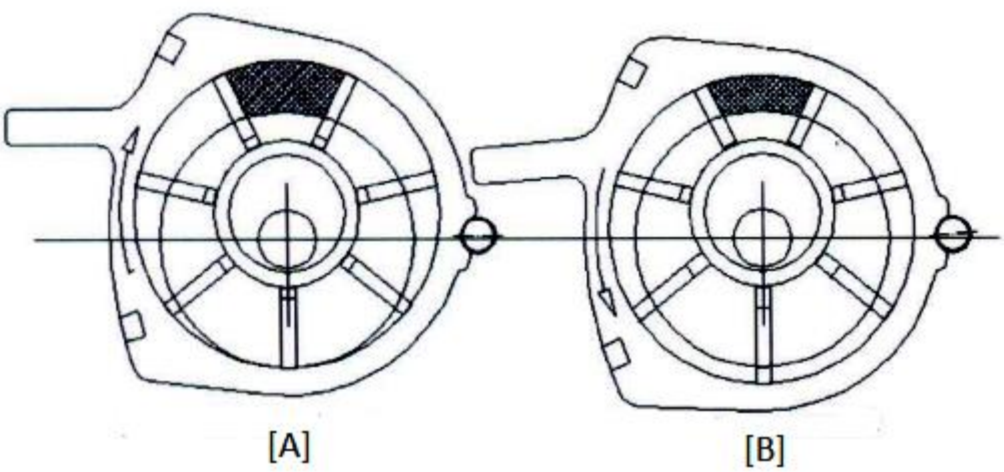


Figure 25. Representation of outer ring in different positions. (23)

3.2.2 Variable Displacement – vane oil pump - US2010/0135835 A1

Patent References (24):

- Patent Application Publication
- Name: VARIABLE DISPLACEMENT –VANE OIL PUMP
- Inventor: Giacomo Arménio, Livorno (IT); Massimiliano Lazzerini, Cenina (IT); Nicola Novi, Pisa (IT); Raffaele Squarcini, Livorno (IT);
- Pub. No.: US2010/0135835 A1
- Pub- Date: JUN. 03, 2010

The concept, as in the previous one, has the basic operating principle of a regular vane pump. The differentiating factor is the method of eccentricity variation, the outer ring is moved in a linear path instead of being rotated around a fixed point. The position control of the outer ring is carried out by the differential of pressure between two chambers, one is assisted by a spring.

Figure 26 represents the concept. The differential between the load applied by spring (5) and the fluid chamber (6) where it is recirculated outlet pressure leads to the linear movement of the outer ring (3). This varies way the eccentricity of the outer ring (3) relatively to the rotor (10), this only has rotational movement.

The vanes (7) located and driven by the rotor (10) and supported by the ring (2) are moved by the driven shaft (1) which drives the rotor (10). The ring (2) function is to ensure a certain position of the vanes when the pump is not working. The rotatory movement of the rotor combined with the oil pressure in the base of the vane (due to oil recirculation to the interior of the rotor) ensure the contact of the vanes with the internal wall of the outer ring (3), the eccentricity ensures the increase of pressure between inlet and outlet ports (7) (8). The (X) represents the rotor axis position and (Y) the outer ring one.

The external ring (3) slides parallel to the spring (5) axis, this way when the spring is in the rearmost position the ring is in the most near concentric position (X coincident with Y) which leads to a lower displacement per rotation, when the spring is in the most stretched position (within the system limits), the design is in the most eccentric position (represented in the figure).

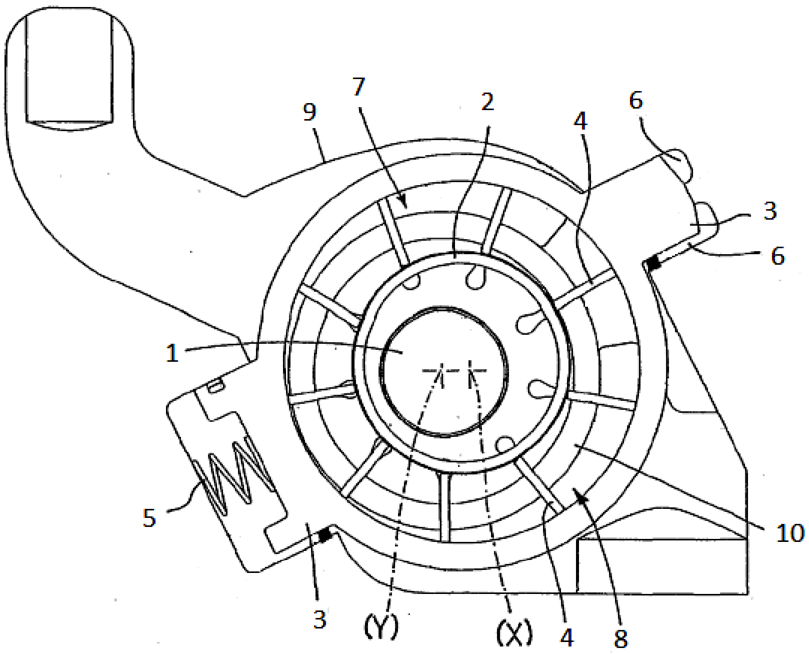


Figure 26. Scheme of variable displacement oil pump with outer ring with linear movement. (24)

3.2.3. Variable Capacity Gerotor Pump - US 7,832,997 B2

Patent References (25):

- Patent Application Publication
- Name: VARIABLE CAPACITY GEROTOR PUMP
- Inventor: Matthew Williamson, Richmond Hill (CA); David R. Shulver, Richmond Hill (CA);
- Pub. No.: US 7,832,997 B2
- Pub- Date: Nov. 16, 2010

The concept has the operating principle of a regular gerotor pump. The variable displacement is achieved by the dislocation along the axis of the external gear (located in the interior of the system), this way the area of contact between internal and external gear is changed.

Figure **27**, A and B, represents schematically the pump concept in two different positions (different displacement per rotation). In

Figure **27** A the pump is in the lower displacement position (lower value of contact area), in B is represented the higher displacement position. **(25)**

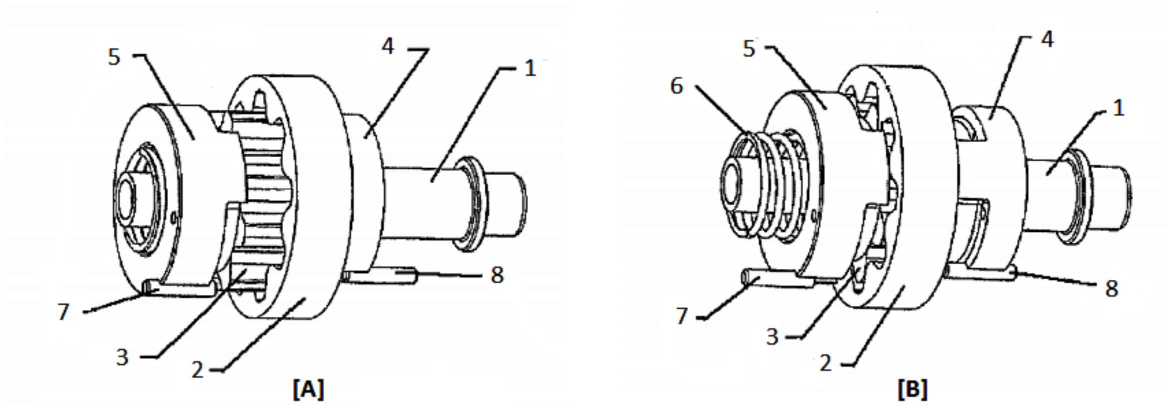


Figure 27. Variable Displacement gerotor Pump Scheme in two different positions. (25)

The system consists of the two usual internal and external gears (2) and (3), however there are two additional elements to the system, a passive and an active pistons (4) and (5), these have linear movement along the driving shaft (1). The movement of the active piston (5) forces the external gear to move, the passive piston (supported in the gear) moves with this one in order to ensure the system sealing and is supported in the spring (6). The movement is dependent on the differential between the load applied by the spring and the fluid pressure on the opposite side (applied at the wall of the passive piston). The pins (7) and (8) ensure the position of the pistons (avoiding rotation of these ones)

Figure 28 represents the system in two different operating positions A and B, which correspond to higher and lower displacement, respectively. The fluid is recirculated from the outlet zone to the chamber (9), which creates a load in the passive piston, this pressure is counterbalanced by the spring (6) load on the active piston, the balance between the positions puts the pump in a higher or lower displacement setup.

Figure 28 (A) is possible to observe that there is full engagement between the gears (2) and (3) and so higher contact area which leads to a higher displacement per rotation. In Figure 28 (B) the opposite occurs.

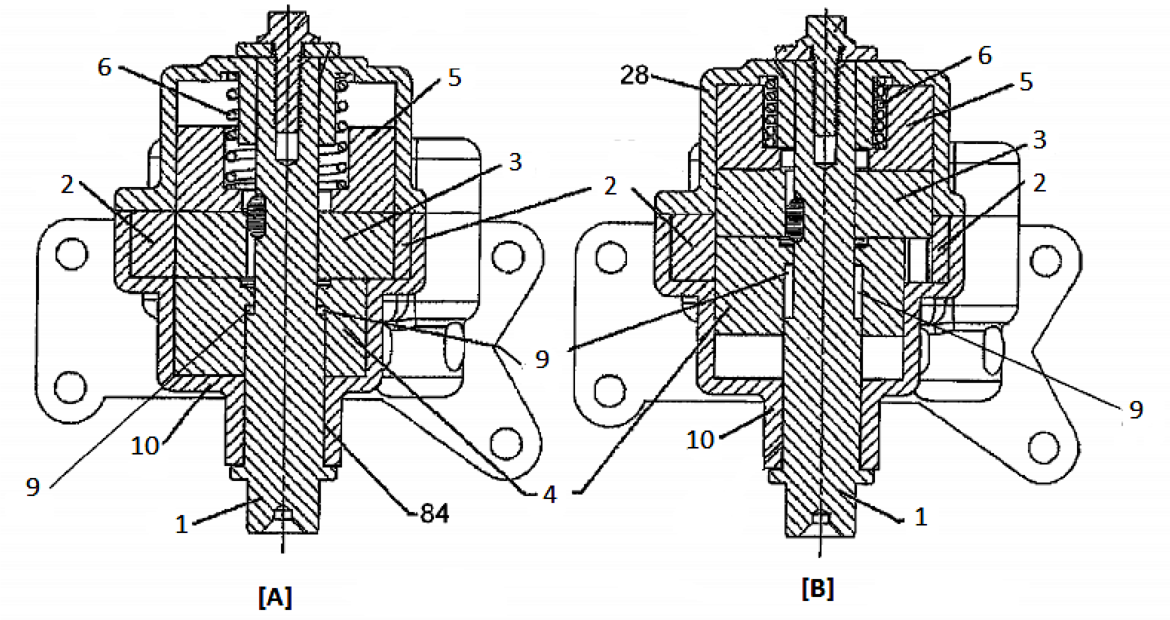


Figure 28. Illustration of the system in two different working positions (higher and lower displacement)

3.2.4. Variable capacity gear pump with pressure balance for transverse forces – US 4740142

Patent References (26):

- Patent Application Publication
- Name: VARIABLE CAPACITY GEAR PUMP WITH PRESSURE BALANCE FOR TRANSVERSE FORCES
- Inventor: Hans. Gunther Rohs, Rechberghausen; Ulrich Rohs, Duren; Jochen Reimann, Eburonenstr;
- Pub. No.: US 4740142

- Pub- Date: April 26, 1998

The concept consists in the operating principle of a regular external gear pump, with two external gears in an outer body which allows the fluid movement from a lower pressure state to a higher (outlet) state. The system is represented in Figure 29, Figure 30 and Figure 31. The variable displacement is achieved by shifting one of the gear (3) in an linear movement parallel to the driving shaft axis (1), this way the engagement area is changed increasing or decreasing the displacement per rotation (a higher area value corresponds to higher displacement).

In Figure 29 the system is represented in a maximum displacement position once there is full engagement between the gears (2) and (3), in Figure 30 the spring (5) is in a rearmost position which leads to a lower contact area between the gears and consequently a lower fluid output per rotation.

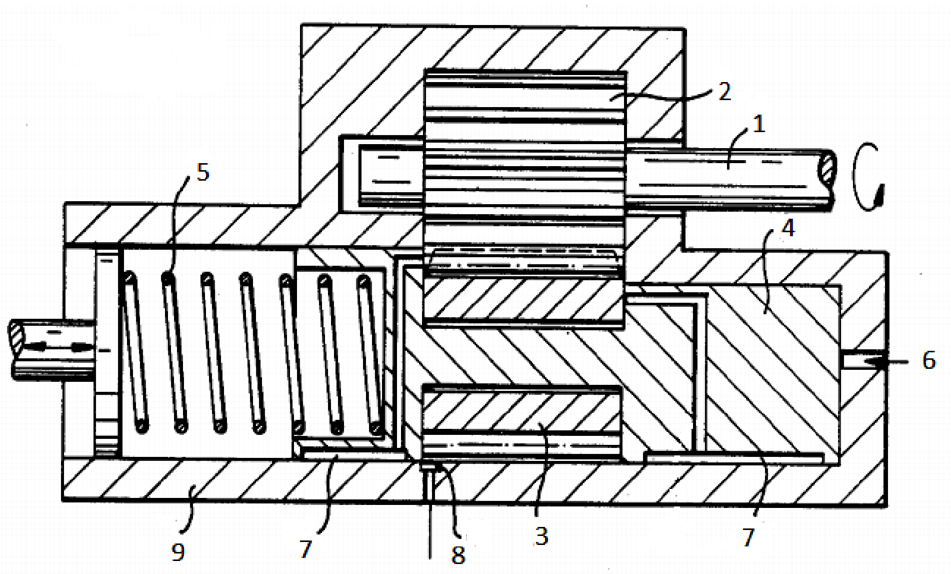


Figure 29. Concept scheme with pump in higher displacement state. (26)

The gear (3) (which possesses the movement capacity position), is defined by the balance between the load applied by spring (5) at one side and the pressure applied on the opposite side by fluid recirculated from the outlet port (11) through channel (6) forcing the piston (4) that is connected to the gear (3) which is driven by gear (2). The piston (4) ensures the system sealing. The channels (7) allow a discharge through (8) in case of over-pressure.

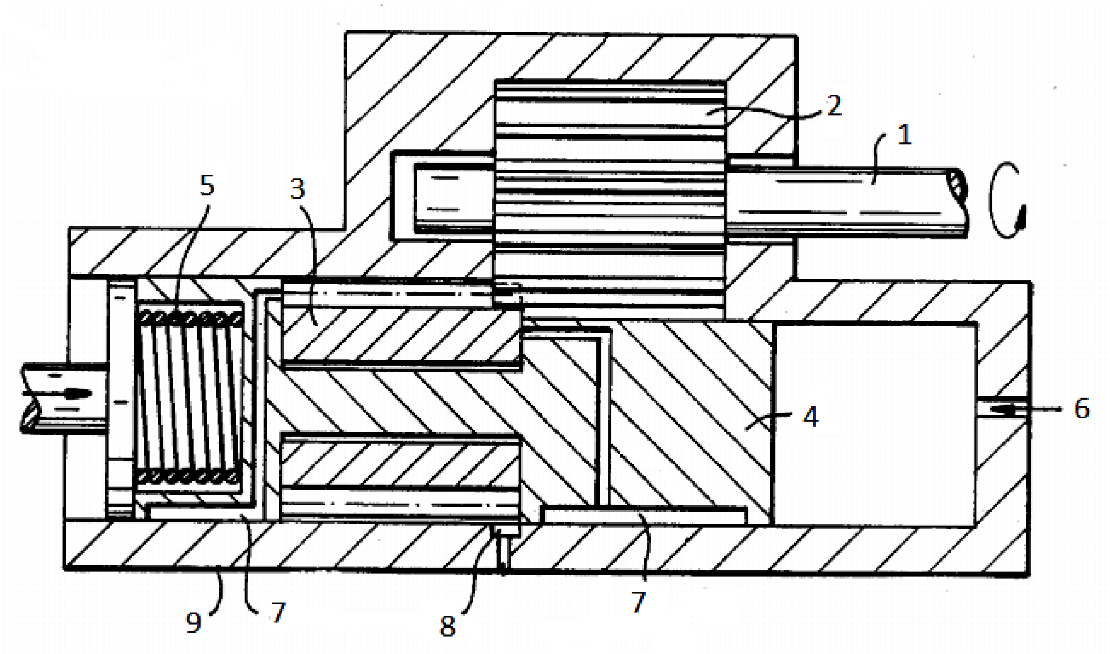


Figure 30. Concept scheme with pump in lower displacement state. (26)

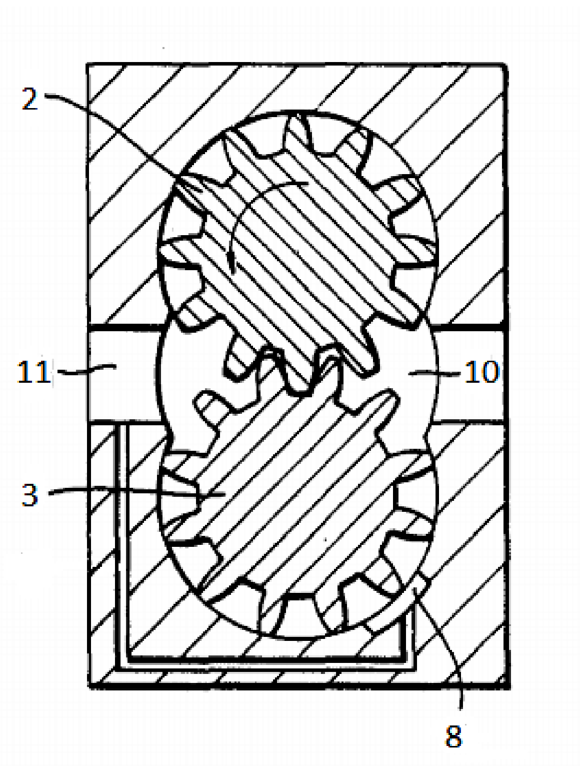


Figure 31. Scheme frontal view representation of concept.(26)

3.2.5 Variable Displacement radial piston pump – US 2006/0222512 A1

Patent References (27):

- Patent Application Publication
- Name: VARIABLE DISPLACEMENT RADIAL PISTON PUMP
- Inventor: Lowell dean Hansen, Brandon, MS (US)
- Pub. No.: US 2006/0222512 A1
- Pub- Date: October 5, 2006

Similarly to the variable displacement vane pumps which ensure the displacement variation through variation of the eccentricity between two parts, the principle applied to this concept with radial piston pumps is the same. The central body which transports the cylinders block is similar to a regular one (for fixed displacement piston pump).

Figure 32 represents schematically the concept. The system is driven by the shaft (1) which transports the cylinders block/rotor (3) where the pistons (5) are located. Due to the centrifugal force caused by the block rotation, the pistons are dragged outside, ensuring the contact with the ring (7). The volume contained in the cylinders (4) is expanding in the zone (Y), inlet, and being compressed in the (X) zone, outlet. The fluid suction and discharge into the system is carried through the cavities (6) in the base of the cylinders.

The variable displacement is achieved varying the eccentricity between the rotor (3) and the outer ring where the pistons are supported (7). This variation is controlled through the balance between the load applied by spring (9) and the piston (10) position, this piston position is dependent on the pressure in chamber (11) where outlet pressure fluid is recirculated into. Both piston (10) and spring (9) are mounted in the body (8) where the ring (7) is supported. This body (8) which is the one with movement rotates in a fixed point (12). When the spring is in a rearmost position, as represented in

Figure 32, the system is in the maximum displacement design position once the inlet volume per rotation is higher due to the longer distance travel by the pistons in the Y (inlet) zone.

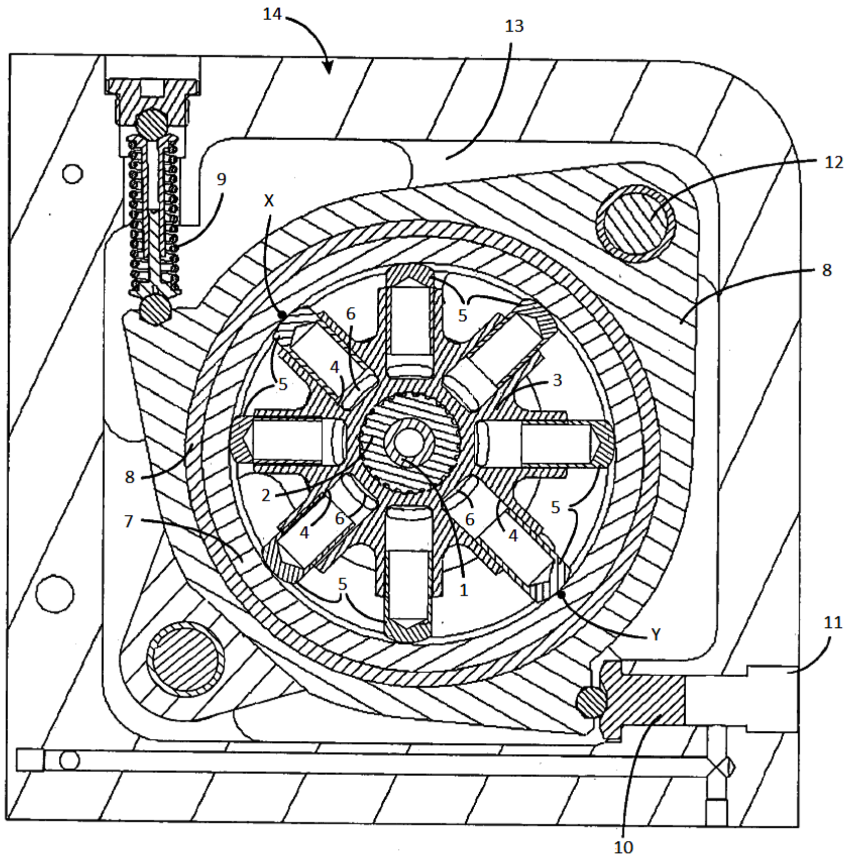


Figure 32. Scheme of variable displacement radial piston pump concept. (27)

3.2.6. High pressure variable displacement piston pump – US 7,887,302 B2

Patent References (28):

- Patent Application Publication
- Name: HIGH PRESSURE VARIABLE DISPLACEMETN PISTON PUMP
- Inventor: Walter Decania Hutto Jr., Cincinnatim OH (US);
- Pub. No.: US 7,887,302 B2
- Pub- Date: Feb. 15, 2011

The concept is based in a regular axial piston pump, the variable displacement is achieved by changing the position of the cylinders block relatively to the pistons, this way the volume dragged per rotation is changed. The Figure 33, Figure 34 and Figure 35 represent schematically the concept, in a frontal view, maximum and minimum displacement positions respectively. The system is driven by the shaft (1) which through the wobble-plate (2) which is disposed in an angle (A) drives the pistons (3) that are moved inside the cylinders chamber (5). The fluid is sucked into the pump through the inlet port (7) and remains in the chamber (6) until being dragged into the cylinders (5) through the channels (6A) and (6B). The fluid is compressed by the pistons thrust and expelled through outlet port (8). The described operating sequence is common to a fixed displacement axial piston pump.

The variable displacement is achieved by the differential factor of the cylinders chamber movement between the positions represented in Figure 34vand Figure 35. The position control of the cylinders chamber is carried out by the pressure difference between the chambers (6) and (7). The channels (16) and (10) conduct the fluid to the controller (EHSV) which leads to pressure variation in the chamber (13) that leads to the movement of the element (5), fluid pressure from the inlet and outlet zones is also applied in opposite sides of the element (12) to assist the movement. In an initial phase of the fluid compression, a certain volume is directed to chamber (6) through channels (6A), when the pistons (3) pass through that channel, the remaining fluid is expelled to (9). This way the cylinders block position variation will depend on the volume that goes in the chamber (6). In Figure 34 does not occur recirculation due to the position of the cylinders, the channels (6A) is sealed in this position and then is ensured the maximum displacement per rotation.

In Figure 35 the cylinders block is in an more advanced position, an so, the channels (6A) are in an totally opened position which means that a higher percentage of the initially sucked fluid is recirculated to the chamber (6) instead of being pressurized and expelled, the system in this position delivers the minimum displacement per rotation.

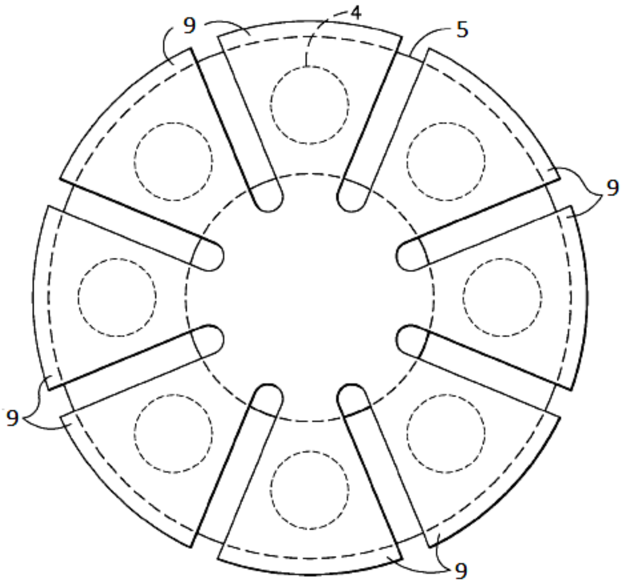


Figure 33. Frontal View of Axial Piston pump scheme. (28)

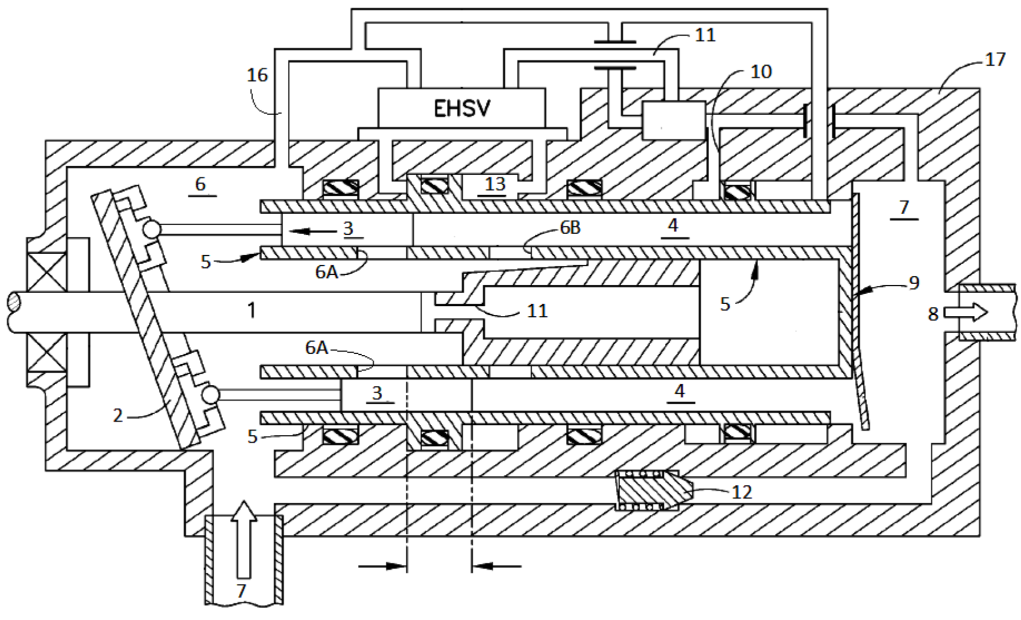


Figure 34. Variable displacement pump concept scheme in maximum flow position. (28)

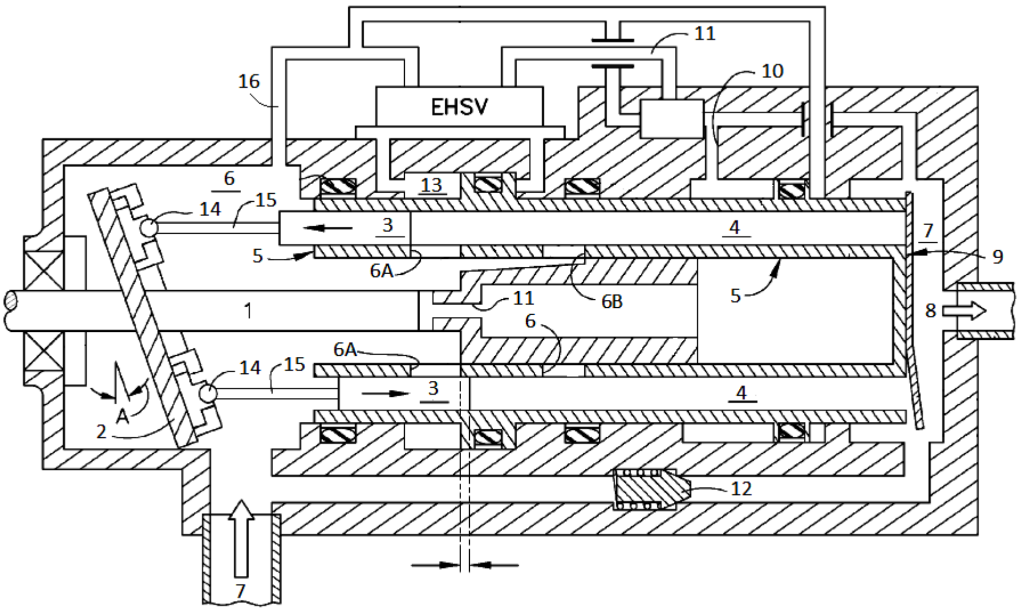


Figure 35. Variable displacement pump concept scheme in minimum flow position. (28)

3.3 Identification of client/market needs

3.3.1 General Process Description

One of the fundamental elements in the development of a product is the identification of the client/market needs, this is essential once the degree of satisfaction of the client will influence the product success and the price that the final user is disposed to pay for it.(29, 30)

Customer satisfaction is the ultimate objective of every business: not to supply, not to sell, not to service, but to satisfy the needs that drive costumers to do business.

The steps that are followed in this process have as objective:

- Ensure that the development process has the focus and is based in the real client needs.
- Identify adjacent needs (not directly identified by the client, but necessary to ensure the primary needs).

- Justify the product specifications.
- Acquire a global knowledge of the needs that need to be fulfilled in an initial phase of the project.

In order to ensure the mentioned requisitions, a systematic and well defined sequence of steps must be followed, this process ensures the focus in the factors that are identified as essentials and an attractive factor for the product success.

- Identify the main factors that need to be considered.
- Collect a general overview of the potential clients (interviews, questionnaires)
- Interpret the collected data (identify the indirect needs).
- Rank the different needs in terms of relative importance to the product and between each other.
- Obtain conclusions on the collected data.

3.3.2 Identification of the main factors/needs to consider

- A - To be 'Environment Friendly'.
- B - Ensure reduction of fuel consumption.
- C - To be reliable and long lasting.
- D - Have low maintenance costs.
- E - Be easily repaired/replaced.
- F - Have a low prize or one that ensures compensation in the long term.
- G - Ensure the reduction of number of oil changes.
- H - Improve the overall performance of the car.

After the identification of such needs, these were classified. In order to ensure that the classification is impartial and based in the market/potential clients opinion, a questionnaire made. The questionnaire is in Attachment A.

The questionnaire was directed to potential clients (holders of a car and driving license), the general results obtained are presented in the

Figure **36**, in this was requested to the person answering the same to select between the eight available options (needs) the three that consider of more importance. In necessary to refer that the questionnaire is set in a fair simplistic way, once the target public however possessors of a vehicle usually don't have any technical knowledge about such specific themes.

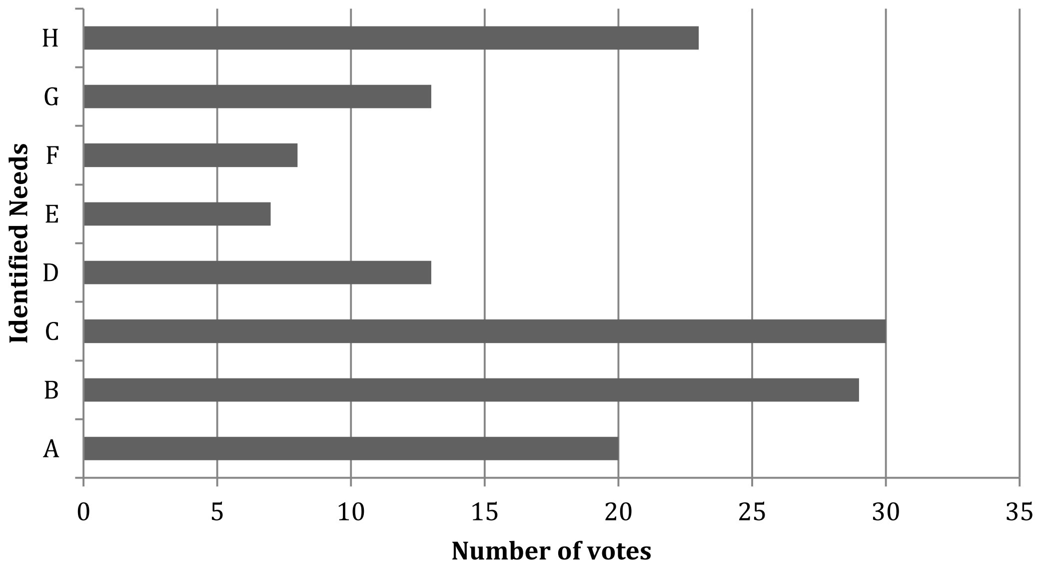


Figure 36. Graphic of the questionnaire results

The analysis of the graphic allows a general overview of the needs that are seen as most important by the general public.

3.3.3 Requirements identification

The main requirements are identified from the information collected in the needs analysis and classification, a need can be translated in several different requirements, Table 3 translates that conversion.

Table 3. Requirements identification

Needs	Associated Requirements
A - To be 'Environment Friendly'.	- Reduction of fuel consumption - Reduction of CO_2 emissions
B - Ensure reduction of fuel consumption.	- Weight (to be light) - Able to respond to flow rate variation needs
C - To be reliable and long lasting.	- Robust - Mechanical elements not subjected to severe wear (number of moving parts)
D - Have low maintenance costs.	- Robust - Mechanical elements not subjected to severe wear (number of moving parts)
E - Be easily repaired/replaced.	- Dimension (adequate to the application) - Weight -Be easily to repair/replace (mechanical connections)
F - Have a low prize or one that ensures compensation in the long term.	- Able to respond to flow rate variation needs - Reduce general costs (maintenance, oil changes, etc.)
G - Ensure the reduction of number of oil changes.	- Able to respond to flow rate variation needs - Reduction of oil cycles - Reduction of oil changes needed
H - Improve the overall performance of the car.	- Weight - Reduce the power required to operate - Able to respond to flow rate variation needs

3.3.4 KANO Diagram

The Kano Model (Kano et al. (1984)), are distinguished three different types of requirements that influence the client satisfaction in different ways. The basic or minimum requirements, referred as ‘Must Be’ correspond to the ones that if not fulfilled by the product will represent an overall dissatisfaction of the costumer, however, at the same time, these are not seen as factors that increase the client satisfaction once they are took for granted, for this reason these must be the basis for the product development.

The requirements that are not expected by the costumer are the ones the contribute for the satisfaction level, these are seen as a sales argument factor are usually classified as ‘One-Dimensional Requirements’. The client satisfaction relatively to the product is directly proportional to the fulfilment of these requirements.

At last, the requirements that are above the expectations of the consumer are referred as ‘Attractive Requirements’, these represent the strongest sales argument and are the ones with more influence in the level of satisfaction. However if this requirements are not reflected in the product that does not bring dissatisfaction once these are above expectations and are not required by the costumer. (31, 32)

The Kano Diagram concept is explained in Figure 37, Figure 38 represents the obtained Kano Diagram.

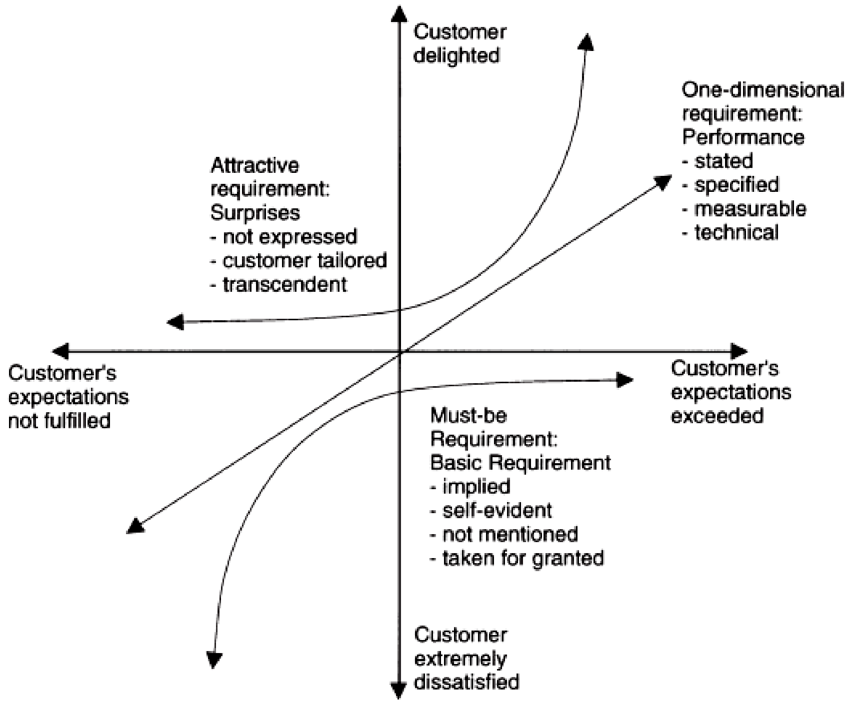


Figure 37. Explanatory scheme of the Kano Diagram. (33)

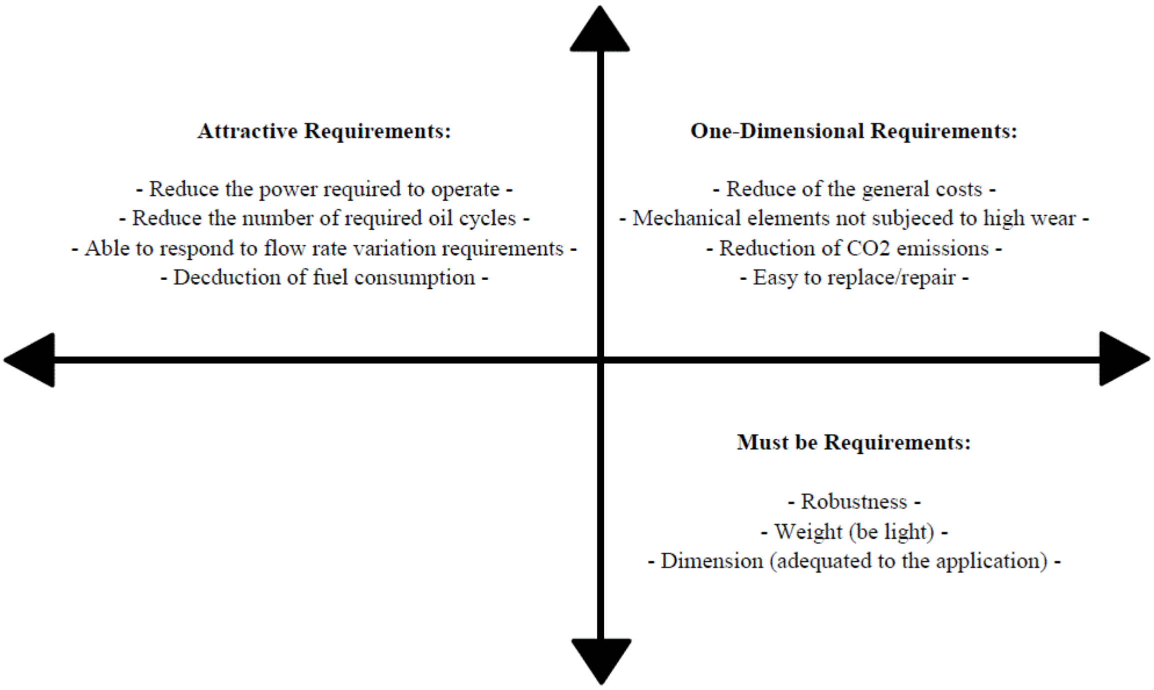


Figure 38. Kano Diagram

3.3.5 Mudge Diagram

In order to attribute an importance scale to the several requirements identified is necessary to do a relative importance analysis. The Mudge Diagram, Figure 39, is a quantitative evaluation of the relation between each two requirements importance.(29, 34)

	B	C	D	E	F	G	H	I	K	K	TOTAL	DEGREE OF IMPORTANCE
A	2	1	2	1	1	1	1	1	2	2	3	3
	B	1	0	1	0	2	1	1	0	0	8	4
		C	2	0	1	0	1	1	2	2	0	2
			D	2	1	2	1	0	0	1	10	4
				E	1	0	0	0	1	2	1	2
					F	1	1	0	1	1	5	3
						G	0	1	2	2	0	1
							H	2	2	2	1	2
								I	1	1	6	4
									J	1	11	5
										K	15	5

Figure 39. Mudge Diagram

From the obtained ranking values, the requirements were divided in five different groups, the five groups represent the classification level of the requirement (from 1 to 5), being 5 the higher importance rate.

<u>Classification Level 1</u>:

- G – Dimension

<u>Classification Level 2</u>:

- E – Robustness
- C – weight
- H – Easy to replace/repair

<u>Classification Level 3</u>:

- A – Decrease CO_2 emissions

- F – Low mechanical wear of the components

Classification Level 4:

- D – Able to respond to flow rate variation requirements
- I – Reduce general costs / improve overall performance
- B – Decrease fuel consumption

Classification Level 5:

- J – Decrease the number of fluid/oil cycles
- K – Decrease the power required to operate the system

3.4 Identification of product specifications

After the identification of the requirements, these need to be translated into specifications, meaning that need to be brought into the product physical level. The specifications must be measurable.

The following specifications were identified:

- Product weight
- Product general Dimensions
- Volumetric Efficiency
- CO_2 emissions level
- Number of limit Km for oil change
- Range of flow delivery
- Range of pressure values
- Oil viscosity (acceptable values for the system)
- System rigidity
- Range of oil temperature values
- Binary
- Power

- Fuel consumption values
- Limit price

3.5 QFD – Quality Function Deployment

In order to develop the product focused in the necessary specification without losing the target to fulfil the costumer/market requirements, is necessary to use a specific tool that allows the correlation/balance between requirements and specifications in a systematic way, that tool is the QFD (Quality Function Deployment) matrix.(35). The QFD method was developed by the Japanese Professor Akao. The QFD Matrix is at Attachment B.

Applying this method, the balance between requirements and specifications is ensured, and so, the technology that is the base of the specifications is used for the customer requirements development. The American Supplier Institute defines QDF as follows:

"A system for translating consumer requirements into appropriate company requirements at every stage, from research through product design and development, to manufacture, distribution, installation and marketing, sales and service."

The QFD Matrix definition goes through the following steps:

- Correlations Matrix: correlation between specifications.
- Relations Matrix: correlation between requirements and defined specifications.
- Product Planning Matrix: Market evaluation, sales arguments identification, specifications to be improved.
- Definition of specifications target values.
- Evaluation of the market similar specifications.
- Evaluation of the technical difficulty to achieve the specifications target values.
- Obtaining conclusions regarding:
  - Critical points (requirements/specification below the market values)
  - Conflict points (impossibility/difficulty of archive two different specification in the same desirable level)
  - Non-satisfied requirements.

- Specifications over the required/necessary.
- Market opportunities.

3.5.1 QFD – Correlations Matrix

The correlations matrix allows the definition of an improvement driver, this means that for each specifications it is indicated if for the defined value its advantageous an increasing, decreasing or if it is indifferent that variation.

Depending on the defined driver it is classified a degree of influence that other specifications have on the one in analysis. Figure 40.

The classification uses a symbolic representation:

Correlation:

- (++) Correlation Positive Strong
- (+) Correlation Positive Weak
- ( ) Correlation inexistent
- (-) Correlation Negative Weak
- (--) Correlation Negative Strong

Improvement Driver:

- ⇧ The higher the value, the better.
- ⇩ The lower the value, the better.
- ⬭ The variation of the values does not have influence.

Improvement Driver	
○	Value variation without influence
▲	The higher the better
▼	The lower the better

Correlation	
++	Positive Strong
+	Positive weak
	Nonexistent
-	Negative weak
- -	Negative Strong

	Weight	General Dimensions	Volumetric Efficiency	CO2 Emmisions Level	Limit Km to oil change	Range of required flow values	Range of Pressure Values	Oil acceptable viscosity	Rigidity	Density	Oil acceptable Temperature Range	Binary/Torque	Power	Fuel Consumption	Limit Price
Weight															
General Dimensions	+														
Volumetric Efficiency															
CO2 Emmisions Level	+		+												
Limit Km to oil change			++												
Range of required flow values			+												
Range of Pressure Values						+									
Oil acceptable viscosity			-			-	--								
Rigidity	-	--													
Density									+						
Oil acceptable Temperature Range							+	+							
Binary/Torque						-									
Power	+			+		-						++			
Fuel Consumption				+								+	++		
Limit Price	-	-	-	-	+						-			+	
Improvement Driver >>>>	▼	▼	▲	▼	▲	○	○	○	▲	▲	○	▼	▼	▼	▼

Figure 40. QFD - Correlations Matrix

3.5.2 QFD – Specification relation to Requirements Matrix

The Relations Matrix translates the importance of each specification relatively to the requirements that these intend to fulfil, as in the Correlations Matrix a classification is assigned, these have three levels:

- 9 – Strong
- 3 – Moderated
- 1 – Weak

Although these are also dependent on other factor, this matrix it is a clear indicator of the importance of each specification for the product. Figure 42.

By the analysis of the matrix, some of the specifications become evident as the more relevant ones:

- Number of Km for oil change.
- Power
- Fuel Consumption
- Binary
- Volumetric Efficiency

Costumer Requirements		Weight	General Dimensions	Volumetric Efficiency	CO2 Emmisions Level	Limit Km to oil change	Range of required flow values	Range of Pressure Values	Oil acceptable viscosity	Rigidity	Density	Oil acceptable Temperature Range	Binary/Torque	Power	Fuel Consumption	Limit Price
Envyronment Friendly	Reduce CO2 Emissions				9									3	3	3
	Reduce Fuel Consumption				1								3	3	9	3
Maintenance	Reduce number of oil changes					9	3					3				3
	Easy to repair/replace	1	1													
Performance	Lower required power				3		3	3					3	9		
	Respond to flow rate variation needs			9		9	1		3							
	Weight (to be light)	9	3							9	9					
	Improve General Performance	3		1	3			1					9	9	3	
Reliability	Low mechanical wear of parts							1	3	3	3	3				3
	Be robust	3	1							9	9					9
	Importance Level	74,9	16,6	158	192	296	107	61,2	80,1	111	111	80,1	159	279	155	202
	Percentual	4	1	8	9	14	5	3	4	5	5	4	8	13	7	10

Figure 41. Specification relation to Requirements Matrix

3.5.3 QFD – Product Planning Matrix

This section of the QFD Matrix consists in an evaluation of the requirements defined relatively to the market direct competitors (equivalent/similar products). In order to have into account the different rank of importance of each requirements, the previous analysis of the Kano and Mudge Diagrams are taken into account, additionally the influence of the requirement as sales argument is also classified. From the comparative analysis to the market value (competition) it is also defined an index of improvement for each requirement, further away from the market value, higher the necessity to improve. The obtained Product Planning Matrix it is represented in Figure 42.

Costumer Requirements		Client	Kano	Grau de importância (geral)	Bechmarking de Mercado: Desirable Product	Bechmarking de Mercado: Market Value		Plan	Sales Argument	Argumento de vendas	Absolute weight	Relative weight
Envyronment Friendly	Reduce CO2 Emissions	3	E	3	4	4		4	1	2	4,5	12,32877
	Reduce Fuel Consumption	4	O	4	5	5		3	1	2	3,6	9,863014
Maintenance	Reduce number of oil changes	5	O	5	5	4		4	1	2	6	16,43836
	Easy to repair/replace	2	E	2	3	3		1	0	1	0,8	2,191781
Performance	Lower required power	5	O	5	4	4		4	1	1	5	13,69863
	Respond to flow rate variation needs	4	O	4	4	5		5	1	1	6	16,43836
	Weight (to be light)	2	E	2	4	4		2	1	1	1	2,739726
	Improve General Performance	4	O	4	5	4		3	1	2	3,6	9,863014
Reliability	Low mechanical wear of parts	3	L	3	4	4		5	1	1	3,8	10,27397
	Be robust	2	L	2	4	3		3	1	2	2,3	6,164384

Figure 42. Product Panning Matrix

3.5.4 QFD – Technical Performance evaluation Matrix

This allows a comparison between the target values defined for each specification and the Market Value.

In this step is defined the target for each technical specification and also referred the market value for each same one in order to have a direct comparison. There is also a parameter that evaluates the technical difficulty implied in achieve the defined value this away it is possible to have a balanced perception of the effort necessary and until which point it is worth it. The matrix is represented in Figure 43.

3.5.5 - QFD – General Conclusions

As referred, one of the most important factors associated to the QFD Matrix is the possibility to obtain at the same time from one global analyses a series of relevant conclusions.

The main conclusions are:

- Critical Points:
    - Reduce the power necessary to operate the mechanism/Pump.
    - Reduce CO2 Emissions (to have a relevant percentage of decrease due to the variable displacement oil pump).
    - Reduce the fuel consumption (ensure that the use of a variable displacement pump contributes for a significant diminution of fuel consumption).

- Market Opportunities:
    - Reduce the number of oil changes, meaning a higher value of specified Km possible without oil change.

- Critical to Ensure:
    - Capacity to properly adapt to required flow changes, this has influence on several requirements as volumetric efficiency, number of oil cycles, etc.

- Over the Market Value:
  - Easy to repair/replace (it can leads to more expensive/complex solution without major return).

		Weight	General Dimensions	Volumetric Efficiency	CO2 Emmisions Level	Limit Km to oil change	Range of required flow values	Range of Pressure Values	Oil acceptable viscosity	Rigidity	Density	Oil acceptable Temperature Range	Binary/Torque	Power	Fuel Consumption	Limit Price
	Unit	kg		%	g(CO2)/km	km (x 100)	L/min	bar	cSt	GPa	g/cm^3	ºC	Nm @ 6000rpm	KW @ 6000rpm	L/(100)km	Euro
Bechmarking	Market value	2,3		70	110	20	10 a 60	<7	12.5 -16.3	26	2,7	70 - 120	1,75	0,6	5	250
	Target Value	<2,5		>80	<114	>35	10 a 60	<7	12.5 -16.3	26	2,7	70 - 120	<1.8	<0.65	<=5	<=250
	Techical Difficulty	2	3	4	5	4	4	4	2	2	2	4	5	5	3	

RP: Product Requirements (Quality Designed / Product Specifications)

Figure 43- Technical Performance Evaluation Matrix

3.6 Concepts Development

3.6.1 Concepts combination table

Taking into account the analysed possibilities of architectures already existent, a concept combination table allows an analysis to cross combination of the main concepts for the different major function of the product to develop.

The Table 4 shows the main considered options for each function.

Table 4. Main Concepts combination Table

Driven	Pump Type	Displacement Variation Method	Control System
Direct	Internal Gear	Rotation in fixed point	Contra pressure (Spring and output pressure)
Off-Axis	External Gears	Linear variation	Electrohydraulic
	Vane	Body Volume Reduction	Electro mechanic
	Pistons		

Based in different combination possibilities, several concepts can be developed respecting the requirements defined as objective for the variable displacement pump.

Although its desirable to develop differentiated concepts, based in the requirements the developed concepts are base in vane pumps once these are the ones that allow a simpler variable displacement system with the desirable specification, this is one of the motive for most of the variable displacement pumps for this type of applications being vane pumps.

3.6.2 Concept 1 – Variable displacement vane pump with outer rotor and side wall movement.

The concept, although based in the usual principle of vane pumps, dragging the volume from a low pressure state to a higher ones, is distinguished by a series of factors as the external rotor and the side moving pump walls. Figure 44 represents the main components essential for the concept development.

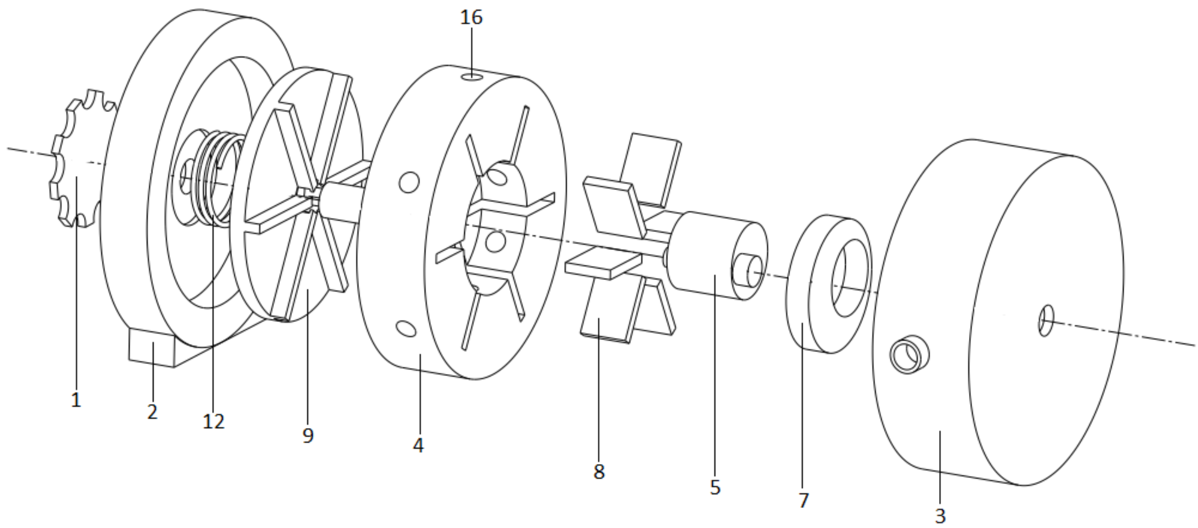


Figure 44. General view of the main components of Concept 1.

One of the most differentiating factor of the concept is the rotational movement of the vanes being driven by the external/outer ring that in this case corresponds to the rotor, the vane are housed in this element. Figure 45 represents a frontal (section) view of the concepts, it's possible with the elements identification to do a general overview of the concept operating principle.

The pump housing/body (2) has two openings, inlet (14) and outlet (10) that through the channels (16) and (4) allows the aspiration and expelling of the oil. The vanes (8), due to the oil pressure in the base (13) are in constant contact with the central element (5). The combination of the vanes rotational movement with the fact of being constantly in contact with the central element (5) ensures the creation of pockets between each two vanes, these are transported through the pump chamber from the inlet to the outlet. The fact of the external rotor being in a decentred position relatively to the central element (5) leads to the fluid pressurization while it is moved to the output.

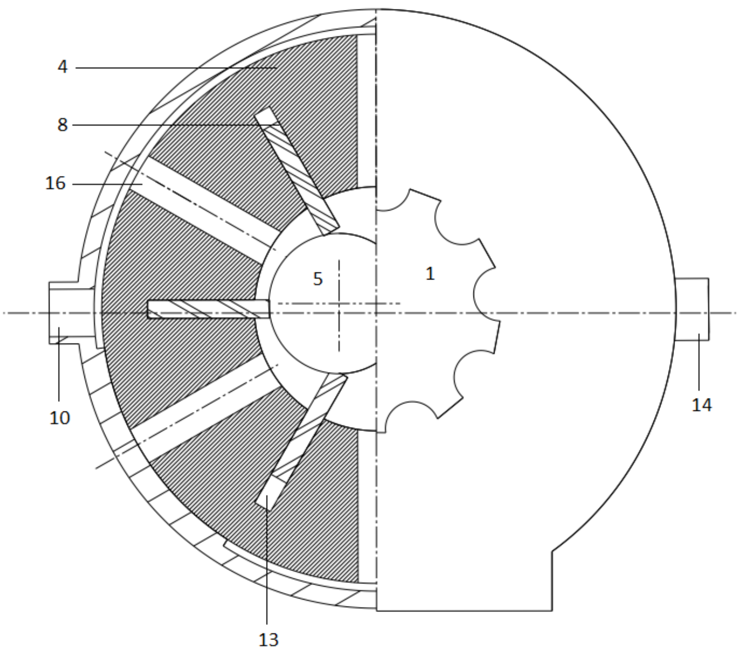


Figure 45. Frontal (cut/section) view of Concept 1.

In Figure 46 and Figure 47 is possible to observe the system at maximum and minimum displacement respectively.

The element (1) drives the rotor (4) which transports the vanes (8). The elements that allow the variable displacement are the sealer (9) and the ring (7), these are moved along the pump shaft axis. The element (7), due to oil pressure recirculated from the outlet zone through channel (6) to the chamber (15), is moved over the part (5), this one does not have rotational movement. The movement of (7) forces the vanes to the enclosure part of the rotor, this way the volume of each pocket is reduced, and so, the displaced volume per rotation decreased. The difference of operating area/length for the two different displacement positions is clearly shown through the figure labels (L1 and L2). As it's possible to observe the longitudinal dimension of the pockets varies from one position to the other, to refer that the eccentric position between rotor and the internal part (5) it is never changed. The position of the vanes is obtained through the balance between the oil pressure applied in (7) and the load created in the sealer (9) by the spring (12). The variation of outlet pressure in (7) is contradicted by the pre-forced spring, and so, for higher pressures the ring (7) will force the vanes into the rotor enclosure part reducing the pockets dimension and consequently the displacement per rotation.

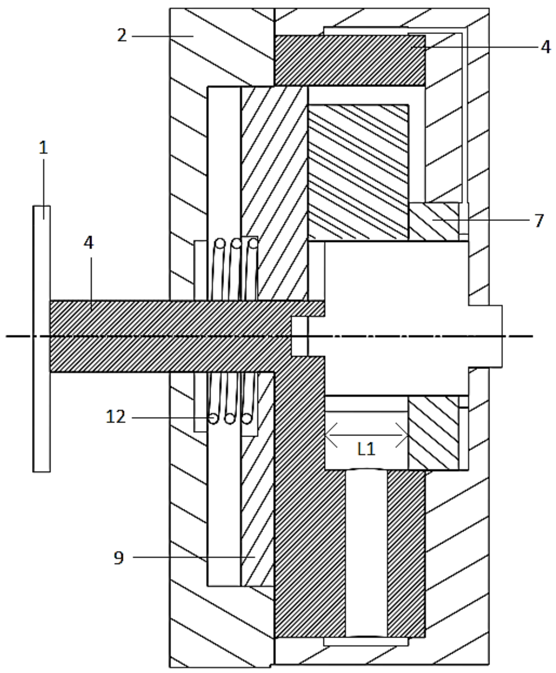


Figure 46. Pump Concept 1 section view at maximum displacement position.

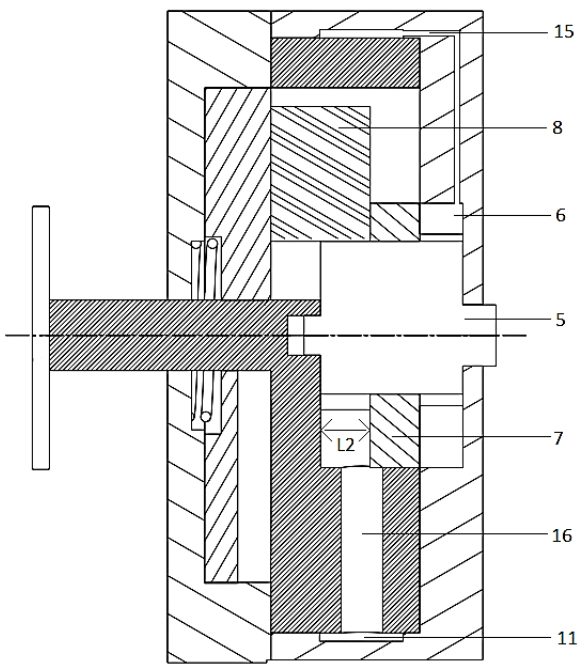


Figure 47. Pump Concept 1 section view at minimum displacement position.

3.6.3 Concept 2 – Variable displacement vane pump with internal rotor and side wall movement

The concept, similarly to the previous one is based in the same principle of a regular vane pump. The variable displacement is also achieved with the movement of side walls that force the vanes to a movement along the pump axis.

Figure 48 represents a model of the basic components that integrate up the concept. Analysing Figure 49, which represents a frontal sectioned view of the pump concept it is possible to have a general overview of the concept basic operating principle. The vanes (5) are driven by the rotor (4) which is mounted in a decentred position relatively to the pump body (outer ring) (3). The vane are in contact with the internal wall of the pump housing (3) due to the combination of centrifugal force (caused by the rotational movement) and the oil pressure applied in the base of the vanes in (6). The combination of eccentricity with the rotor movement ensures the transportation of oil from the inlet zone (13) to the outlet (11), the pocket (between each two vanes) dimension variation along the route, due to the decentred position it is ensured the oil pressurization.

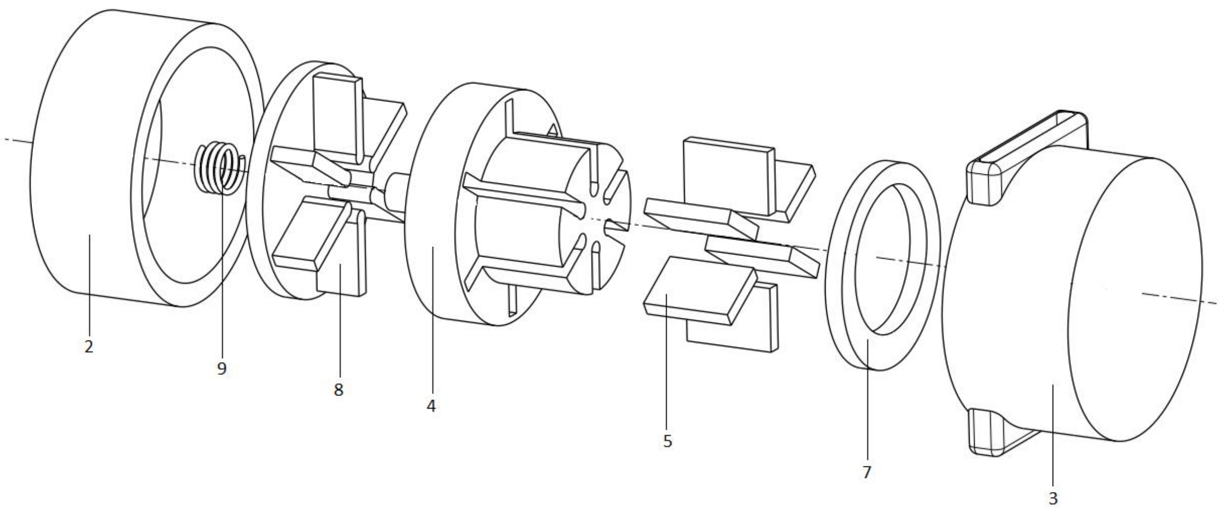


Figure 48. General view of the main components of Concept 2.

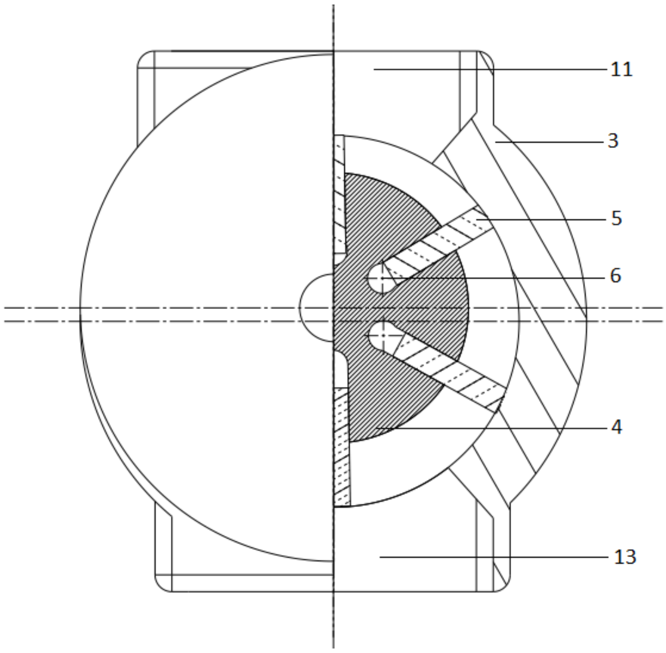


Figure 49. Frontal (cut/section) view of Concept 2.

Figure 50 and Figure 51 show the section view of the concept in a maximum and minimum displacement positions respectively.

The variable displacement is reached through the elements (7) and (8) movement. The ring (7) is forced in the side wall of the vanes due to oil pressure in the opposite side, the oil is at outlet pressure once is recirculated from the outlet through channel (12) to the chamber (12). The movement is linear and occurs on the top of the rotor, the ring does not have rotational movement. The ring forces the vanes to the enclosured part of the rotor where these (the part of the vanes enclosured) do not have functional action. The variation between two different operating positions is shown in the figures through the indications L1 and L2.

The position of the vanes at each moment is controlled by the balance between the oil pressure load in (7) and the contradicting force of spring (9) pre-forced in the sealer (8), the sealer is positioned in the opposite vanes side wall of the ring (7). The sealer (8) rotates jointly with the rotor (4).

This way as it is possible to observe, in figure Figure 50 the major area of the vanes is in contact with internal wall of the body pump and not collected in the rotor, corresponding this to a maximum displacement position, the opposite is represented in Figure 51.

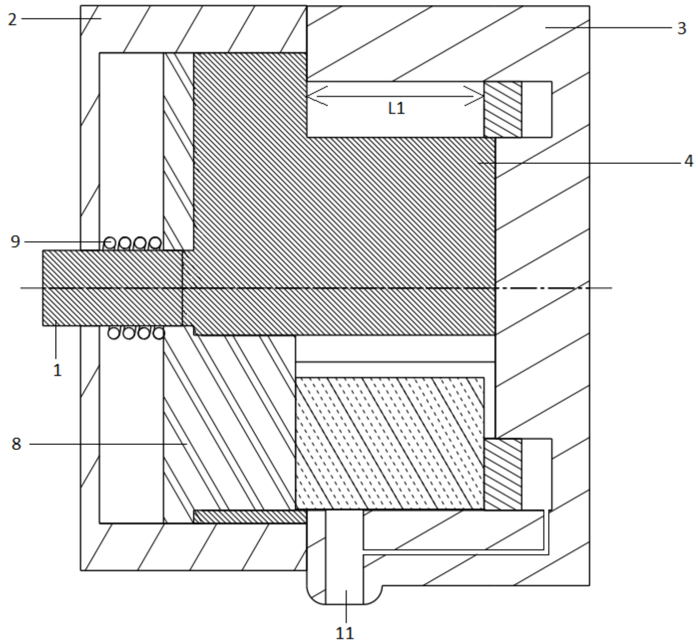


Figure 50. Concept 2 section view at maximum displacement position.

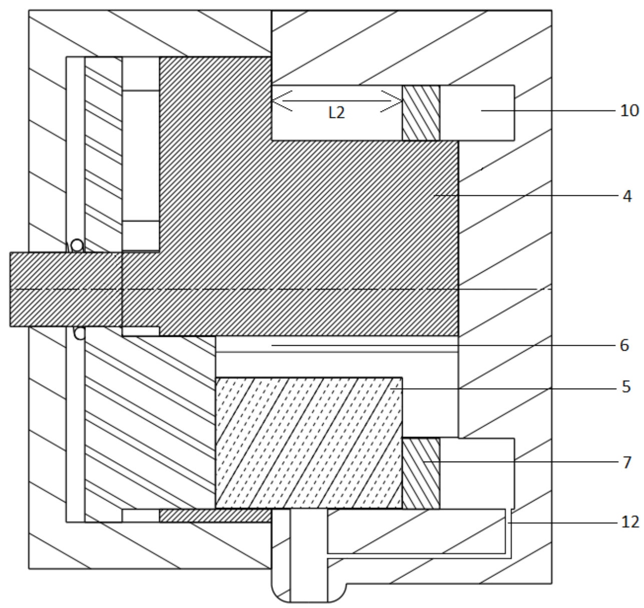


Figure 51. Concept 2 section view at minimum displacement position.

3.6.4 Concept 3 – Variable displacement vane pump with variation by steps

The concept is based in a regular vane pump. The differentiating factor is in the partitioned vanes that allow the variable displacement. Contrary to what happens in the previous concepts, only one element possesses linear movement. This part designed combined with its movement involve a certain part of the vanes (partitioned), the involved vanes are not functional, and this way the displacement volume per rotation varies. The vanes are actually a set of components, each vane is composed by three 'sub-vanes'.

Figure 52 presents a three-dimensional representation of the main components that are required to the concept. Analysing Figure 53 is possible to understand the basic operating principle of the concept, also in Figure 54 and Figure 55 the system is represented in the maximum and minimum displacement positions, respectively.

The rotor (1) drives the vanes (4), these are kept in contact with the internal wall of the pump body (stator) (2) due to the combination of centrifugal force caused by the rotational movement and oil pressure in the vanes base, the oil is recirculated from the outlet zone to the chamber (7). The quantity of active sub-vanes at each vane is defined by the position of the parts (5) and (6). Part (6) rotates driven by the rotor (1). Part (6) is forced by the spring (8) on one side, while the part (5) is subjected to oil pressure which contradicts the spring load. This way the parts (5) and (6) will move at the same time along the pump axis. When these move forward the vanes are collected into part (5), this way a certain number of sub-vanes (1 or 2) become not-active. The part (6) purpose is to ensure the vanes position (decentred) when collected into (4).

In Figure 54 it is possible to observe that all the vanes are in contact with the wall of the pump (no vanes collected into the part (5)), this way this corresponds to the maximum displacement position, the opposite is visible in Figure 55, 2 of the 3 sub-vanes are collected in part (5), meaning that there is no contact with the internal wall of the pump body, this is the minimum-displacement position.

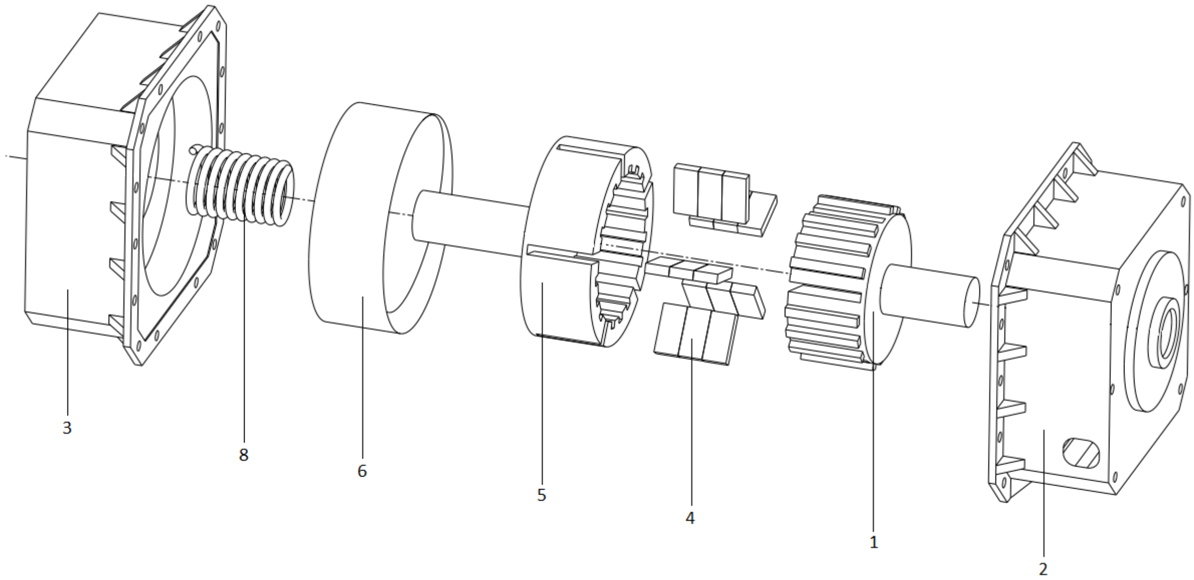


Figure 52. General view of the main components of Concept 3

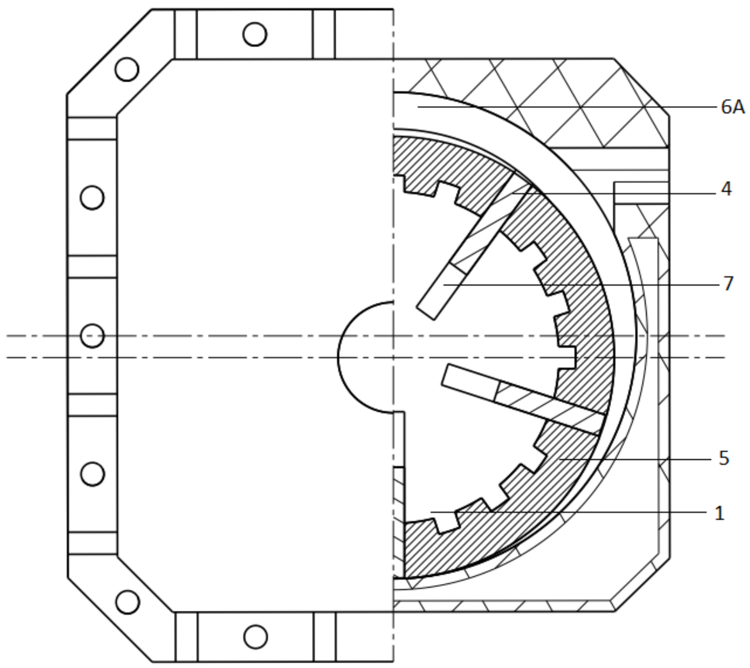


Figure 53. Frontal (cut/section) view of Concept 3.

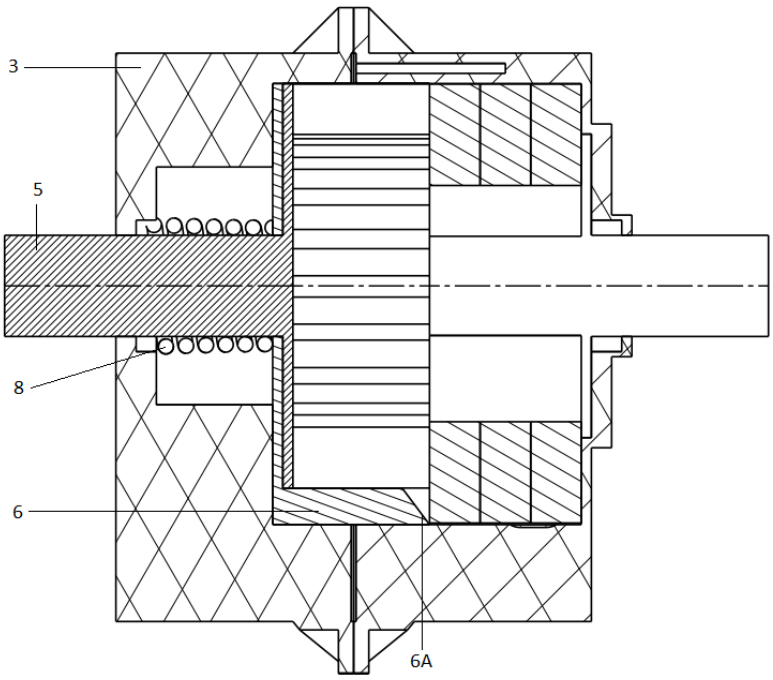


Figure 54. Concept 3 section view at maximum displacement position.

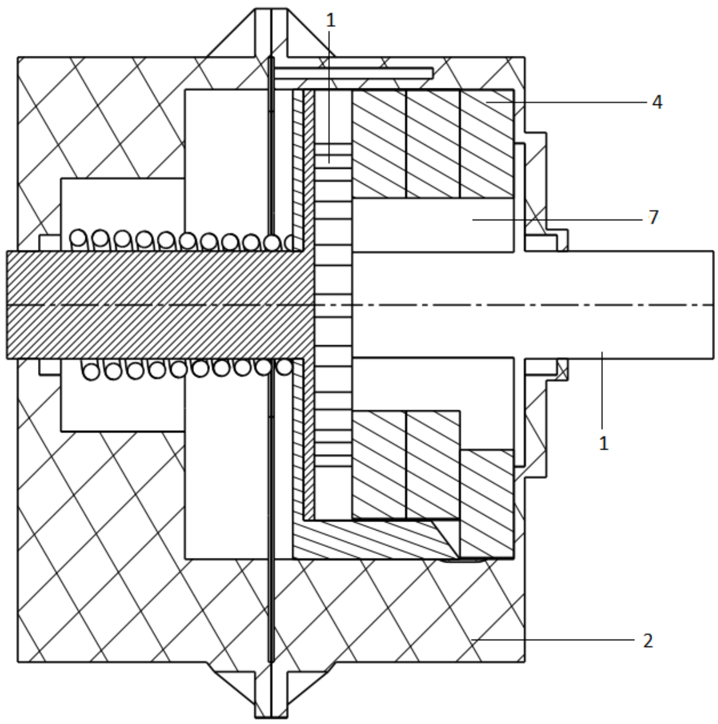


Figure 55. Concept 3 section view at minimum displacement position.

3.6.5 Concept 4 – Variable displacement vane pump with variation through eccentric cylinder movement.

The concept consists in an vane pump where the are driven by an outer rotor, these are in constant contact with an internal element which is in an decentred position relatively to the rotor, this element does not have rotational movement, however it does have a variable position relatively to the rotor (eccentricity variation), is due to this movement that the pump ensures variable displacement capacity. The variation of the position is carried out by a system pin-channel. In Figure 56 is possible to observe a three-dimensional representation of the main components that are required for the concept.

In Figure 57, Figure 58 and Figure 59, it is possible to have a perception of the system basic operating principle.

The rotor (1) which is the driving element of the system transports the vanes (12) which are in constant contact with the central cylindrical body (4) due to the oil pressure applied through the chamber (13), the constant contact ensures the creation of pockets (between each two vanes) that combined with the rotational movement leads to the transfer of oil from the inlet (9) to the outlet, the oil goes in the pumping chamber through the channels (10).

The control of the system eccentricity, which defines the displacement per rotation is led by the components (4) and (5). The control is defined by the angular position of the part (4A) relatively to the pin (5A) which is driven by (5) and drives (4). The linear movement of the part (5) where (5A) is fixed leads to the rotation of (4) which rotates in (4A). The element (4) is connected to (4A) in an eccentric position, this way when (4) rotates in (4A) the position of (4) relatively to the rotor (1) varies. Figure 60 represents the system in two different displacement positions.

The control of element (5) position is defined by the balance between the load applied by the pre-forced spring (7) in (5C) and the oil pressure which is applied in the opposite side (5B) which contradicts the first one.

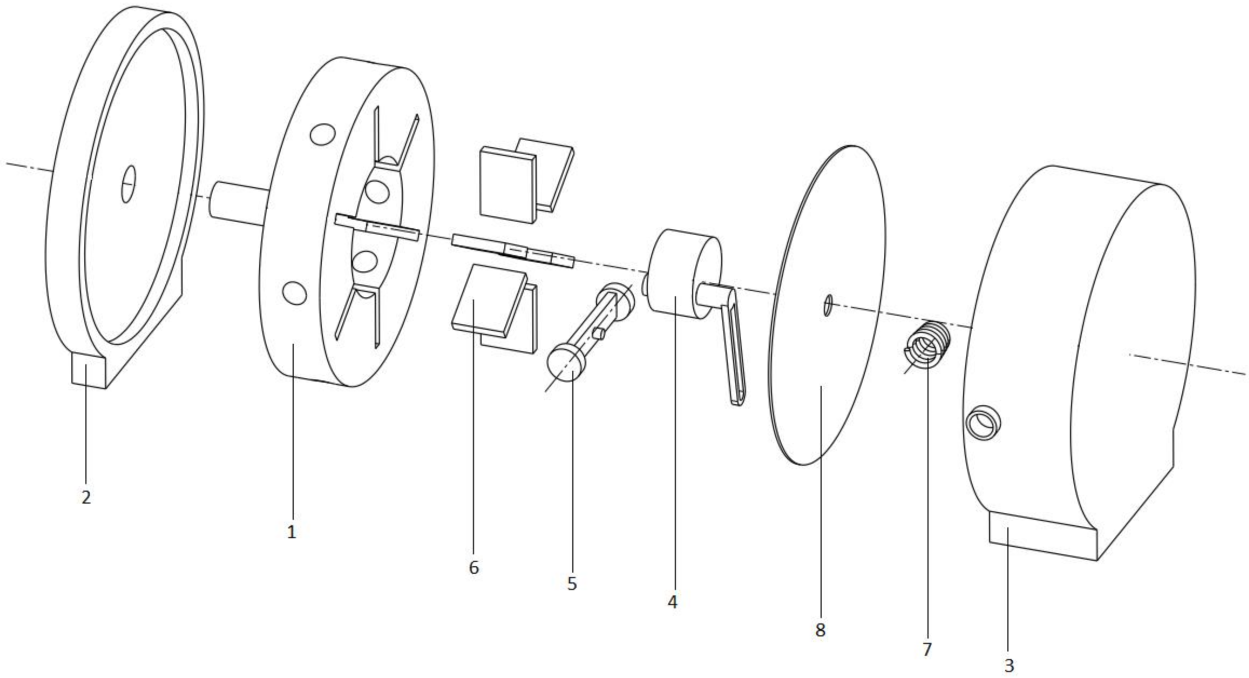


Figure 56. General view of the main components of Concept 4

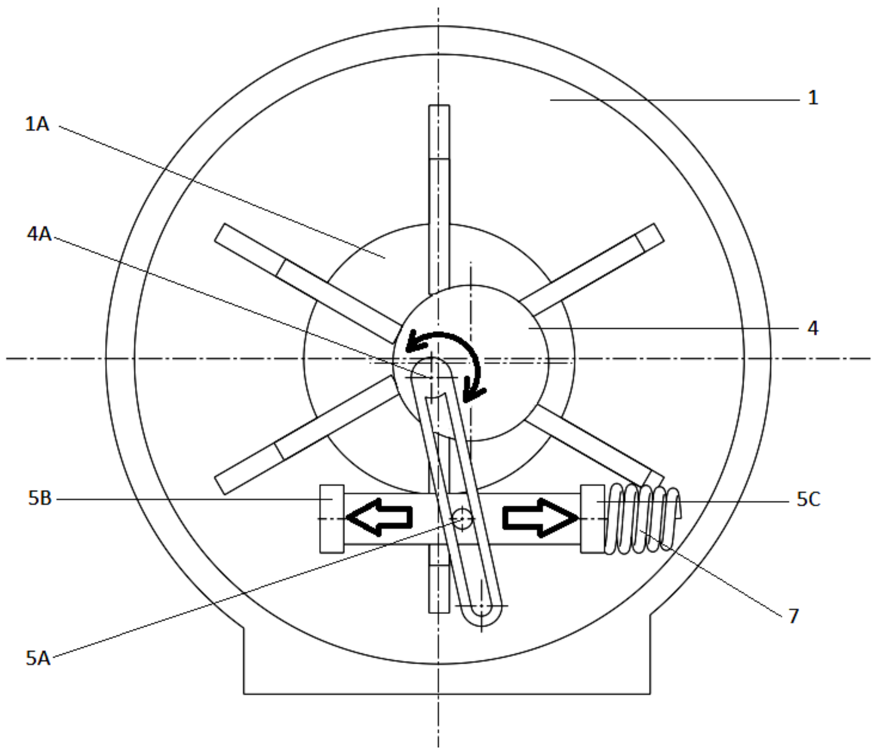


Figure 57. Frontal View of Concept 4

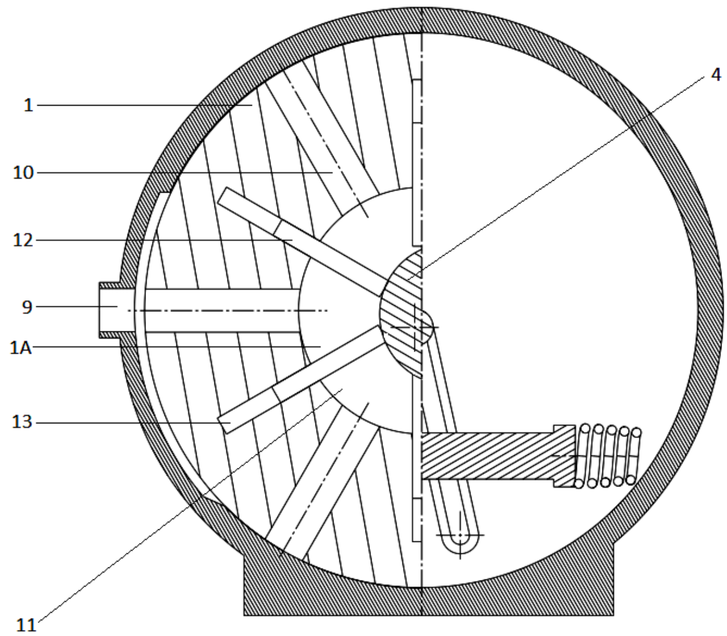


Figure 58. Frontal (section view) of Concept 4

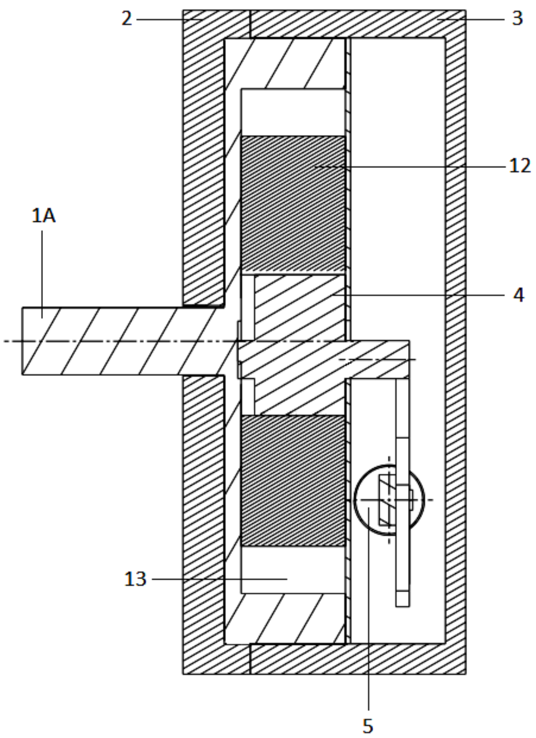


Figure 59. Side (Section view) of Concept 4.

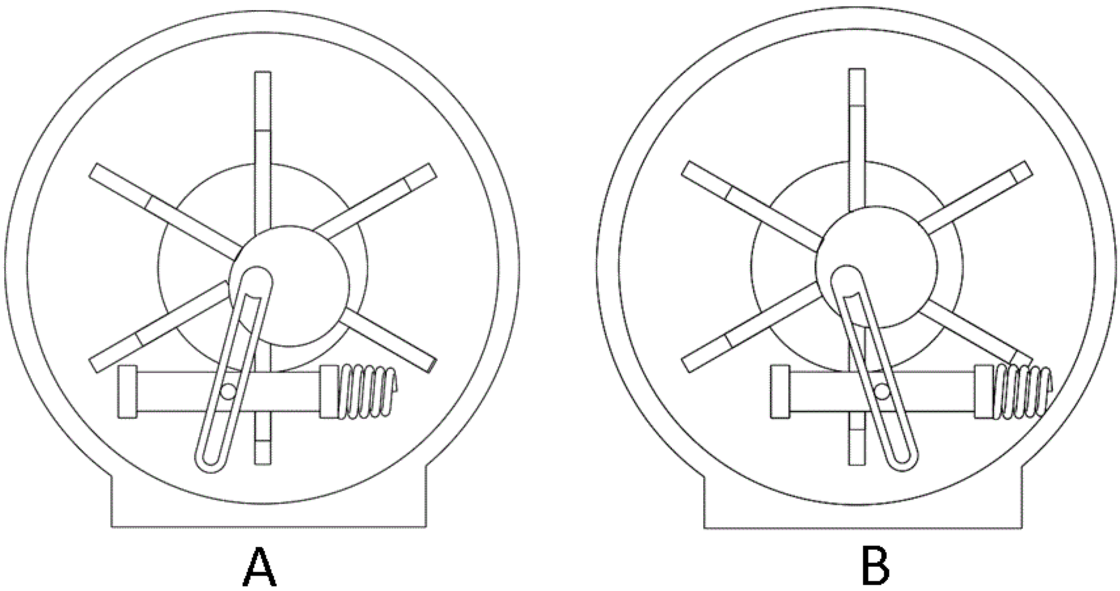


Figure 60. Concept 4 represented in two different displacement positions.

3.7 Concepts Analysis/Selection

After the development of several concepts is necessary to evaluate each one in order to decide the one that is more adequate for the purpose and advance with this for a more complete concept/project.

In order to rate the several concepts, these are evaluated in the more relevant aspects that characterized each one, the main factors considered were:

- Price (Based in the number of main parts needed for the concept)
- Reliability (based in the number of parts and concept complexity)
- Reduction of fuel consumption/CO_2 emissions/Efficiency (based in the system operating principle.
- Difficulty associated to the production process of the concept.
- Innovation level (degree of differentiation relatively to the market existing solutions)

The factors evaluated have different percentage weight values due to the different requirements that are associated to (already ranked). The evaluation of each factor/specification is done relatively to the other according to the scale of **Table 5**.

The concepts classification is represented on Table 6.

Table 5. Concept classification rank reference table.

Relative performance	Factor
Worse than the reference	1
slightly worse than the reference	2
Equal to the reference	3
Better than reference	4
Much better than reference	5

The rank between concepts is then obtained by summing the multiplication of each specification considered classification by its percentage weight as presented in Equation 1.

$$Classification/Rank = \sum_{spec=1}^{n} Pontuation * Percent(\%)$$

Equation 1

The chosen concept is Concept two that ensures the variable displacement through the moving of the vane into the active area or the enclosure area in the rotor. This way this does not need eccentricity variation in order to vary the volume per rotation.

Specification	Concept								
	Weight	Concept 1		Concept 2		Concept 3		Concept 4	
		Classification	Relative Weight	Classification	Relative Weight	Classification	Relative Weight	Classification	Relative Weight
Price	20	3	0.6	4	0.8	2	0.4	3	0.6
Reliability	20	3	0.6	4	0.8	1	0.2	3	0.6
Efficiency	30	3	0.9	4	1.2	1	0.3	4	1.2
manufacturing process	20	3	0.6	4	0.8	3	0.6	3	0.6
Innovation	10	3	0.3	3	0.3	3	0.3	3	0.3
TOTAL		3		3.9		1.8		3.3	
RANKING		**3th**		**1st**		**4th**		**2nd**	

Table 6. Concepts Classification Table

3.8 Product/Concept Architecture

The product architecture is the set of functions that lead to a physical mapping of the product physical elements.(30)

The product architecture can be better defined as:

- The arrangement of physical elements.
- The mapping of physical and functional elements.
- The specification of interfaces between the different components.

In order to facilitate the planned architecture is usual to represent it in a schematic form, this way the Figure 61 represents the overview of the product architecture while Figure 62 represents more specifically the planned architecture in hydraulic scheme.

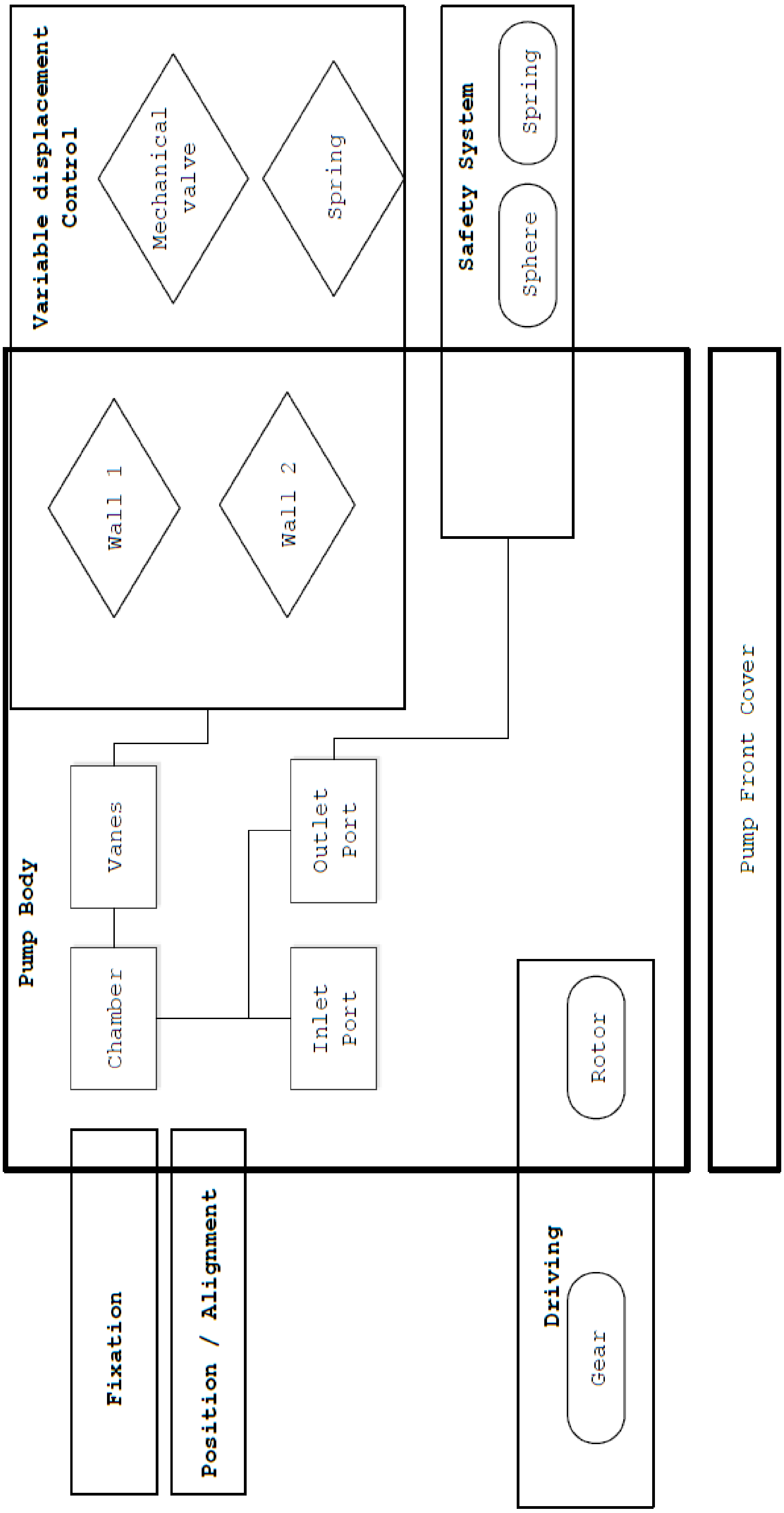


Figure 61. Product Schematic Architecture

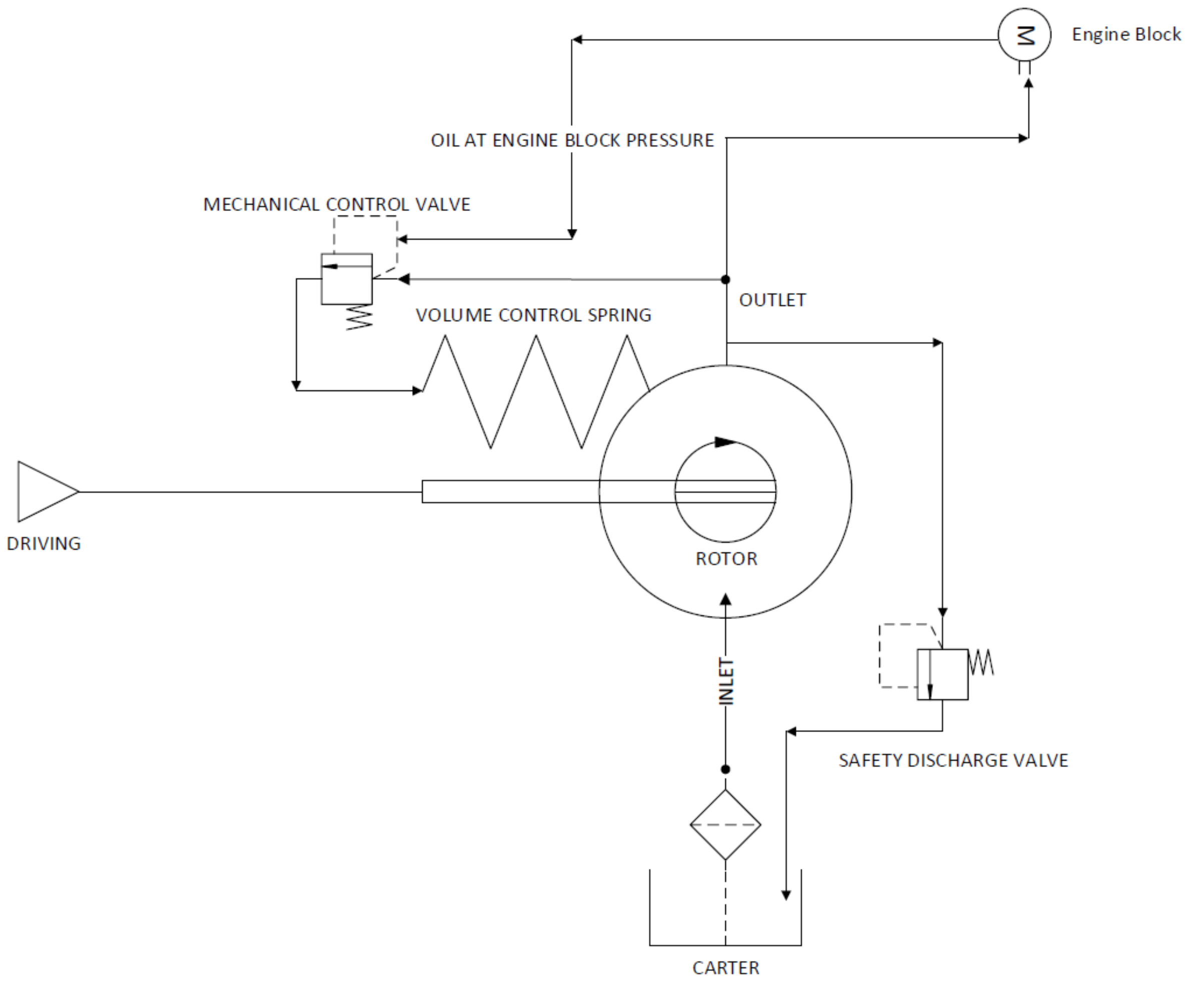


Figure 62. Product main hydraulic architecture representation

Chapter 4

Product Mechanical Design

As referred the chosen concept was Concept 2, a variable displacement vane pump with pumping chamber variation through side wall movement. The movement is controlled by the balance between two chambers, one assisted with a pre-forced spring. The internal rotor transports the vanes, this is driven by the pump shaft. The control chamber is connected to a mechanical control valve whose position is controlled by the balance between control pump chamber and oil pressure from the engine block. There is also the possibility of a hydraulic electro-valve which regulates the control chamber pressure

The main challenge in the design of the concept is that it has to be adapted to systems already existent, makes no sense to develop an oil pump that doesn't fit into the space or with inlet and outlet connections that are not in line with the current market solution. This way, the objective is to develop a different operating system within the same physical space and with the same main interfaces in other to be easily adaptable and appealing to the market. A commercial variable displacement oil pump was taken as reference in order to fulfil the requirements.

The main factors into account:

- Overall dimension of the product.
- Compatible interfaces (inlet and outlet)
- Compatible driving shaft.
- System Internal volume (influence the displacement per rotation) in line with the needs of the main commercial similar systems.

As already referred the chosen concept was Concept 2, besides the limitation referred some characteristics need to be defined at the beginning.

The initial calculation was in order to define the dimensions that the pumping chamber must have in order to fulfil the flow rate requirements. The pump taken as reference (commercial oil pump for 1.5l DCI engine) is a variable displacement with eccentricity variation. In this concept the eccentricity is fixed, based in values for the rotational speed range of the pump were calculated the medium eccentricity value, the biggest pocket area and volume. With these values it’s possible to do a connection to the concept values, the main (bigger) pocket need to ensure the same maximum and minimum volume values at maximum and minimum displacement positions.

The axial movement of the side wall needs to ensure the volume variation that in the commercial pumps it is get through the eccentricity variation. This way the number of vanes (same as number of pockets) was defined as the same for the commercial pumps (7), the eccentricity value (in the concept fixed) was defined as the medium value for the commercial ump in the rotational speed range, the pocket area was defined as the one that minimizes the side wall movement length which is an critical value once it influences the pump dimensions and systems mechanic stability.

Based in this data, the pumping chamber was resized relatively to the commercial reference pump, all the other components were dimensioned in line with the ones present in the commercial pump (vanes, safety discharge valve, control mechanical valve), this represents a challenge due to the existence of an axial movement of the chamber, which does not occur in the commercial pump.

This way an effort was made in order to ensure the correct positioning and dimensioning of all the components within the new concept, ensuring at the same time that the same volumetric characteristics are respected, also, the global dimension of the pump and inlet, outlet and driving axis position are the same as in the commercial pump (better explained ahead)

4.1 Process for pumping chamber dimensioning

As referred the considered values as reference are from a commercial design variable displacement oil pump, dimensions to obtain chamber volume and eccentricity variations were taken in the 3D model of this one.

For initial analysis were considered the reference values for required flow (l/min) at each moment (engine rotational speed [rpm]), in Table 7 are represented some values used as reference (values obtained from the cases analysed in the state of the art review).

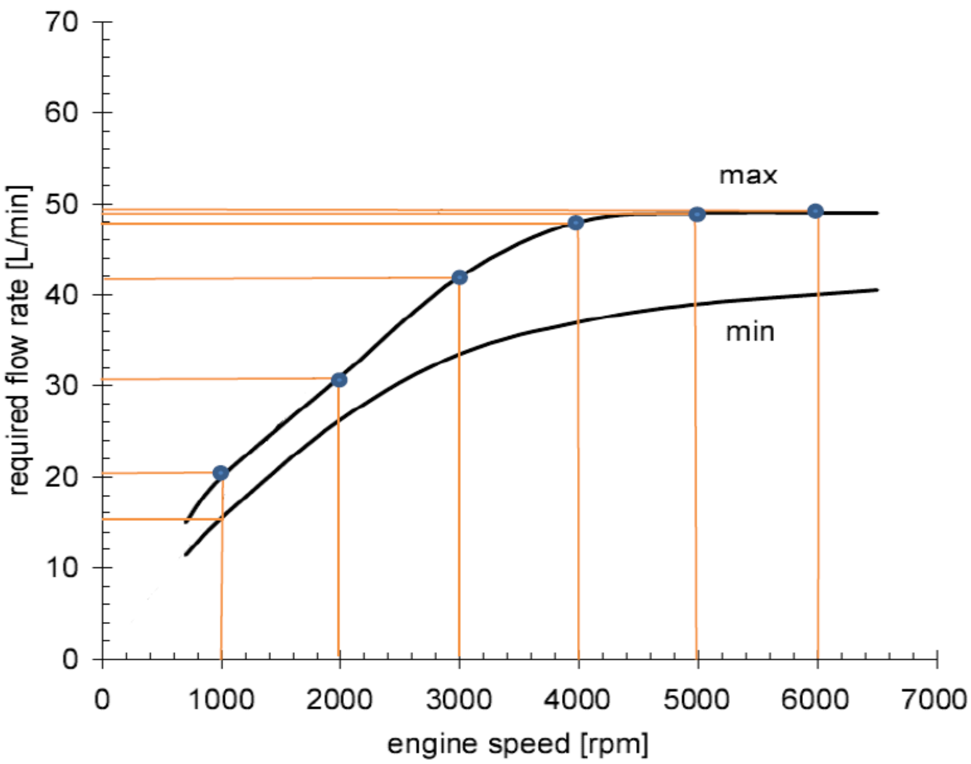


Figure 63. Required Flow Rate at different rotational speeds.

Table 7. Flow vs Rotational Speed (reference values).

Flow [l/min]	Rotational speed [rpm]
15	1000
31	2000
42	3000
50	6000

With the full range flow values at each rotational speed, is possible to obtain the volume per rotation (l/rot) which corresponds to the full volume of the pumping chamber (is an approximate value once does not have into account the losses and recirculated fluid at each rotation).With the values of chamber volume and knowing that this is composed by seven sub-chambers/pockets is calculated the approximated value of volume per pocket (the pockets don't possess all the same volume do to the eccentricity of the system), the interest is to obtain the maximum and minimum volume per pocket in order to ensure that these ones are maintained for the new concept development.

The calculated values for maximum, medium and medium pocket volume are:

- Maximum Volume per pocket=0,00214 [l]
- Minimum Volume per pocket=0,00154 [l]
- Medium Volume per pocket=0,00194 [l]

Once the new concept have variation of the length, this will be a main factor in the determination of the chamber volumes (max and min), in the commercial pump the chamber length is constant (28,35 [mm]), this value must be taken into account in order to don't be largely exceeded in the new concept, once the overall dimensions of the new pump must also be near to the ones of the commercial one. Firstly was calculated the area per pocket which in the commercial pump varies with the eccentricity but in the new concept is fixed and should be calculated in order to optimize the maximum necessary length of the pumping chamber.

Taking into account the considered maximum and minimum volumes per pocket and the length of the commercial pump it's possible to calculate the area for each:

- Maximum Area per pocket=78 [mm^2]
- Minimum Area per pocket=53 [mm^2]
- Medium area per pocket=68 [mm^2]

This values serve as reference to create an iteration in order to find the area per pocket for the new concept (which is fixed), this iteration is executed in order to find the area value that minimizes the chamber length.

The value defined of area per pocket for the concept pump is 82 [mm^2]. With this value, in order to ensure the referred maximum and minimum volume values of 0,00214 [l] and 0,00154 [l], respectively, the maximum length of the pumping chamber is 28,40 [mm] which is near to the commercial pump value of 24,35 [mm]. The minimum length (that ensures the minimum volume value) is 18,46 [mm], this way the chamber length will vary in a maximum of 9,94 [mm].

As referred the eccentricity of the commercial pump is variable (between 0,2 and 2,7 [mm]), for the concept pump this value is fixed in 1,3 [mm] which is near the middle value of the commercial pump, ensuring that the level of compression is near the main requisites and also ensures the space for the movement of sealing parts in the chamber.

For the minimum flow rate delivered by the concept is necessary to consider a minimum rotational speed (1000 [rpm]) and the minimum pumping chamber length, the value achieved is 10,56 [l/min]. Considering the opposite with maximum length (28,40 [mm]]) and a rotational speed of 4500[rpm] (near the value where the flow delivery needs stabilize) the maximum value is of 65.29 [l/min]

Figure 64 represents the obtained concept pumping chamber (only representative), its visible the decentring of the rotor relatively to the chamber housing, the referred (dimensioned) pocket is the one in orange once is the one that represents the pumped volume (both maximum and minimum). The decentring would be ideal if in the recirculation side (in green) the space between rotor and housing were the minimum (near 0 [mm], however it would not be possible to have moving parts between rotor and housing as it is necessary in this concept.

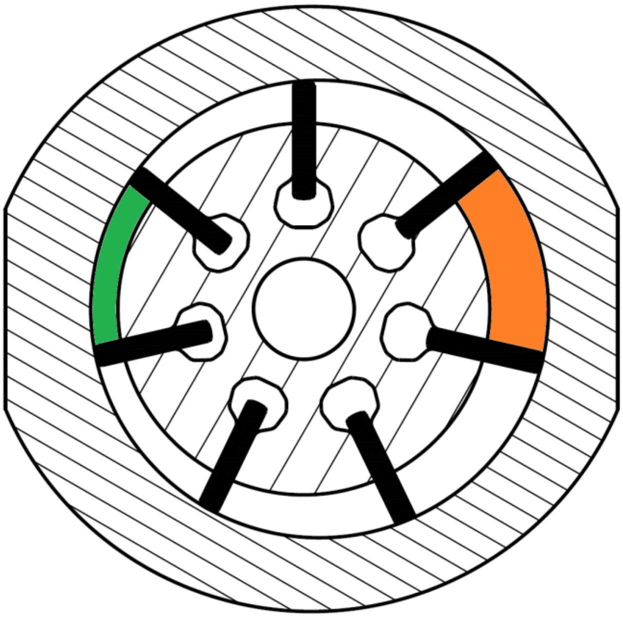


Figure 64. Illustration representative of the pumping chamber

Table 8. Rotational Speed and required flow vs chamber length

Speed (rpm)	Flow (l/min)	Chamber Length (mm)
1000	15	26,13
2000	31	27,00
3000	42	24,39
4000	48	20,90
5000	59	20,55

4.2 Main Parts/Sub-Systems

An overview of the main parts design is done in this sub-chapter, ahead is done a general description of the hole system, based in the obtained concept prototype.

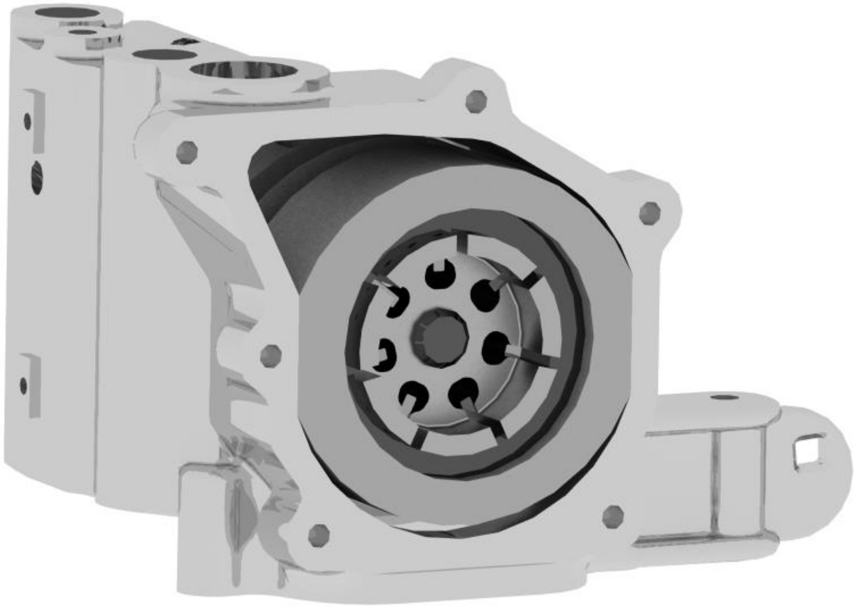


Figure 65. General view of CAD concept pump model.

4.2.1 Main Body

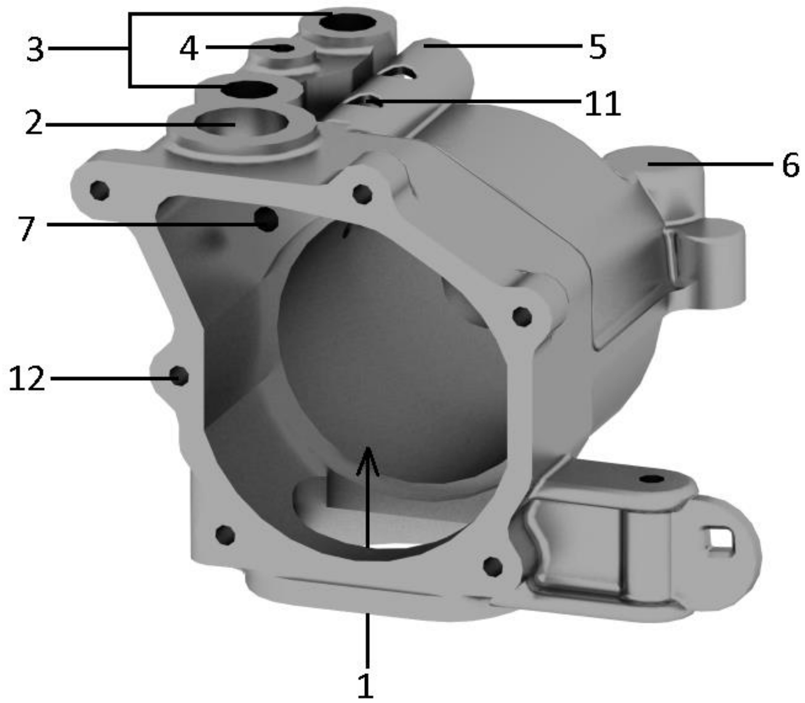


Figure 66. Concept Body, Isometric View.

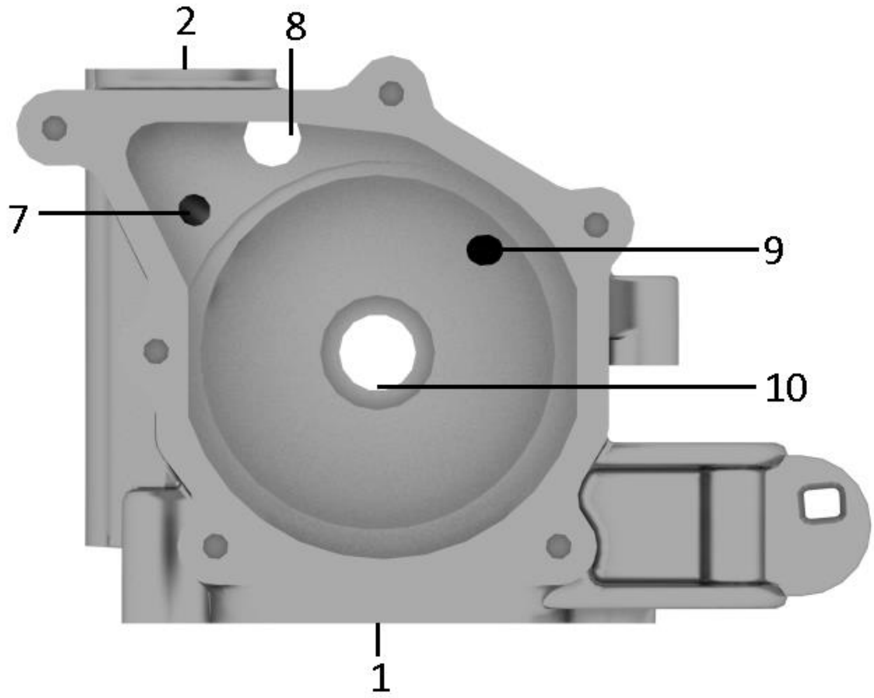


Figure 67. Concept Body, Front View.

In a general overview of the pump main body, this is one of the most important and complex elements of the design. This need to ensure all the parts of the mechanism fixation and interaction and still the alignment with the vehicle chassis and interaction with elements external to the pump.

- 1 – Inlet Port
- 2 – Outlet Port
- 3 – Alignment/Fixation Points
- 4 – Mechanical Control Valve inlet
- 5 – Safety Discharge Valve ‘housing’
- 6 – Electro-Valve Housing
- 7 – Recirculation chamber (from pumping chamber to control chamber)
- 8 – Connection to Safety Discharge Valve
- 9 – Channel to drive fluid from control chamber to electro-valve
- 10 – Rotor Shaft position
- 11 – Holes for Safety valve discharge
- 12 – Threaded holes for Front Cover assembly

4.2.2 Rotor

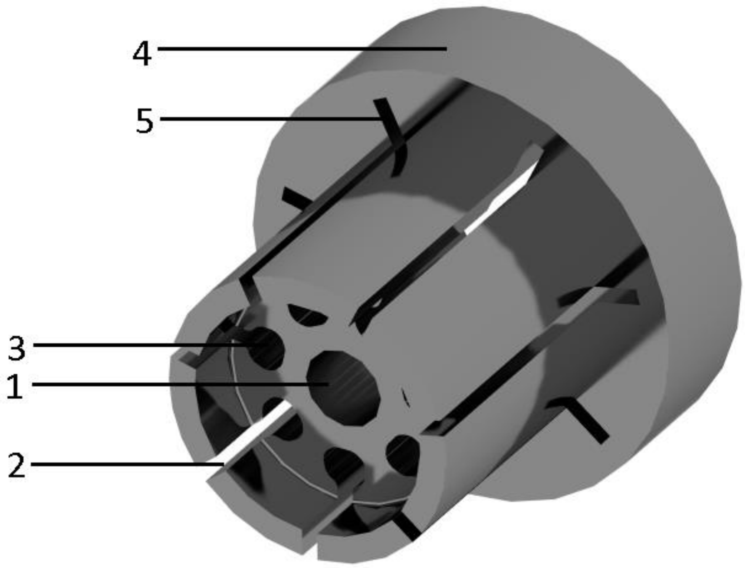


Figure 68. Concept Rotor

The rotor is the element driven by the shaft and responsible to drive the vanes, in this case due to the pump functionality the rotor have an additional area where a certain part of the vanes can be enclosure (4).

- 1 – Driving Shaft Position
- 2 – Vane(s) position – functional/pumping area
- 3 – Chamber applying oil pressure in the vanes.
- 4 – Rotor enclosure area.
- 5 – Vane(s) position – enclosure/non-functional area.

4.2.3 Rotor Sealer

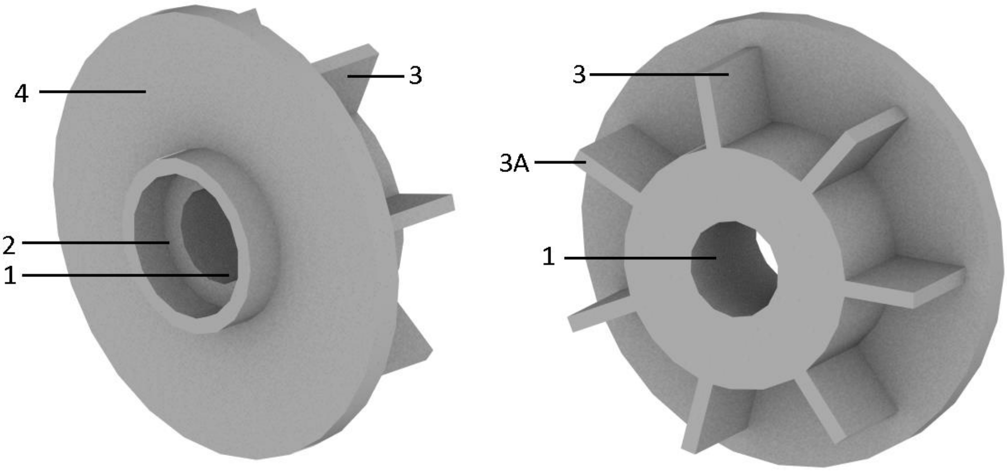


Figure 69. Pump Concept Rotor Sealer

The Rotor Sealer is driven by the rotor itself. It seals the rotor once this is opened along the vanes gaps.

- 1 – Pump driving shaft position.
- 2 – Pre-forced Spring contact area
- 3 - Fillers (go inside the enclosure rotor areas)
- 3A – Frontal fillers surface (in contact with the vanes).
- 4 – Rotor Sealer posterior surface (in contact with the fluid in the control chamber)

This element, in addition to ensure the sealing of the rotor is still one of the parts that guides the vanes position, the fillers are in contact with the vanes in the enclosure rotor area. This element is forced by the spring in the control chamber and also by the oil pressure in this one.

4.2.4 Pump body Front Cover

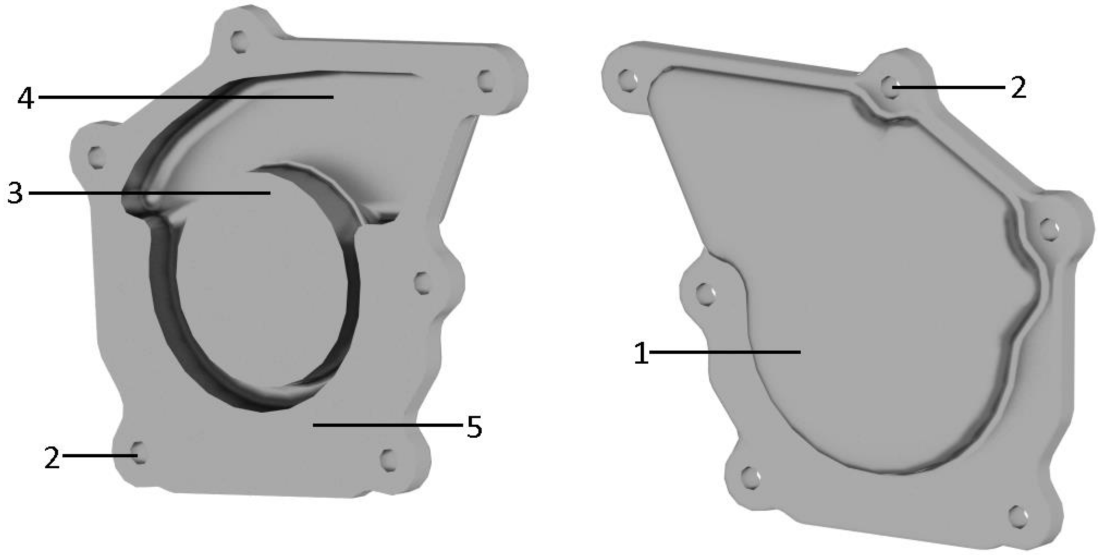


Figure 70. Front Cover

The front cover ensures the sealing of the pump body, this also incorporates a chamber (in the rotor opposite side of the control chamber).

- 1 – Cover external surface.
- 2 – Cover holes for screw tightening with the pump body.
- 3 – Internal Surface (in contact with the rotor)
- 4 – Chamber cavity.
- 5 – Internal Surface (in contact with pump body).

4.2.5 Mechanical Safety Discharge Valve

The safety discharge valve is used in order to ensure that in case of any pump element failure that leads to an uncontrolled increasing pressure in the pumping chamber, the entire pump mechanism is not compromised and catastrophically fails, this way, in case of over-pressure the oil in the pumping chamber is discharged through the safety valve.

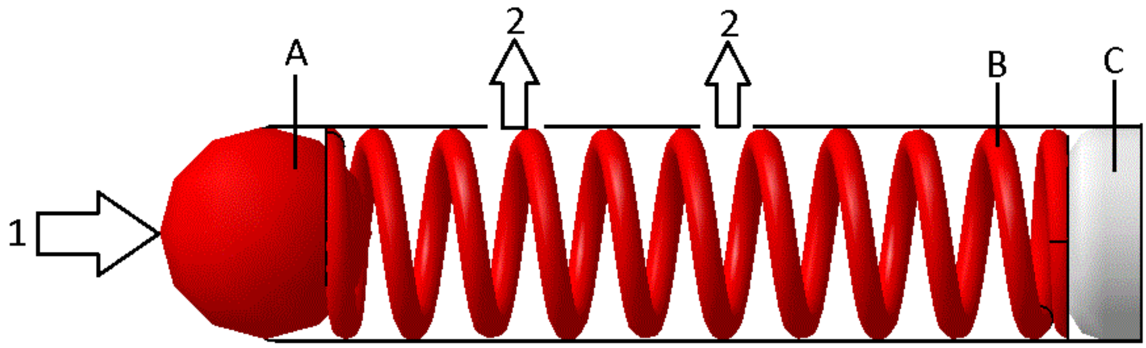


Figure 71. Safety discharge valve scheme

- A – Valve sphere (element where pressure is applied).
- B – Spring that forces the position against the oil pressure ensure that movement only occurs at overpressure situations.
- C – Valve Sealer.
- 1 – Pressure applied by the pumping chamber oil (the valve is located near the pump outlet port).
- 2 – Discharge openings.

4.2.6 Mechanical controlled Valve

The mechanical control valve is controlled by the balance between oil at control chamber pressure and engine block oil pressure. The system is purely mechanical and it is consists in a valve with spring, the valve position variation leads to variation in the control chamber pressure.

- A – Mechanical valve.
- B – Mechanical element that limits the valve course.
- C – Spring that contradicts the valve descending movement.
- D – Valve sealer.
- 1 – Inlet oil from the control chamber
- 2 – Opening to communicate with 3 when the valve in a lower position.

- 3 – Inlet oil pressure from the engine.

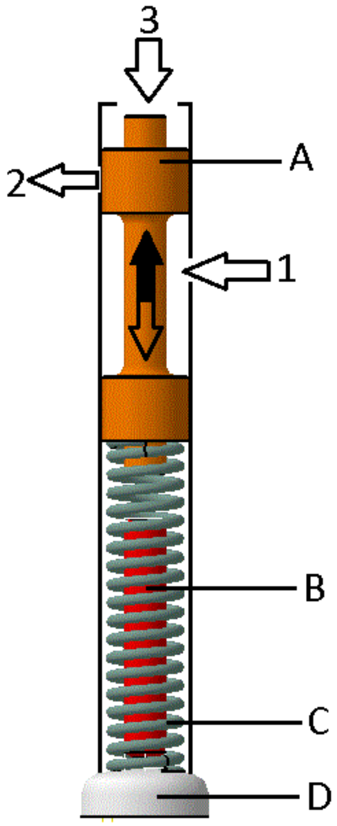


Figure 72. Mechanical valve to pressure control scheme

4.3 Adaptability to a existent/commercial design

As referred the pump was designed in order to easily fit to the existent systems, this means that the main dimensional characteristics of the concept must be similar to an existent one ensuring this way the easy integration in the Engine system.

In Figure 73 to Figure 77 the Concept pump body (Grey body) and commercial pump body (Blue Body) are overlapped in order to allow the visualization of the coincident main parameters:

- Inlet Port Dimensions
- Outlet Port Dimensions
- Alignment Points coincidence

- Dimensions Similarities (the concept 'box' dimensions do not exceed the ones from the commercial pump)
- Driving Shaft Position Coincidence.

These elements coincidence represent a big effort in the development of the concept once a different operating principle for the variable displacement is used.

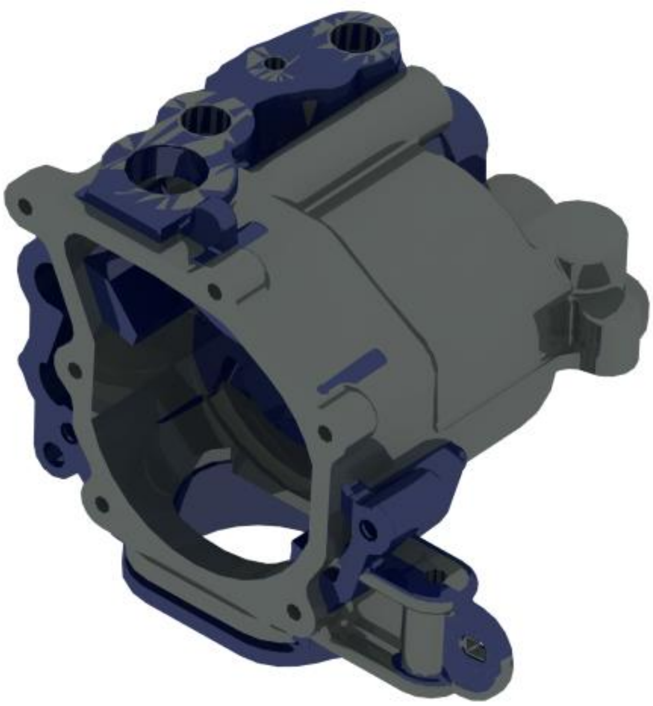


Figure 73. Concept and Commercial Pump Mainbody overlapping.

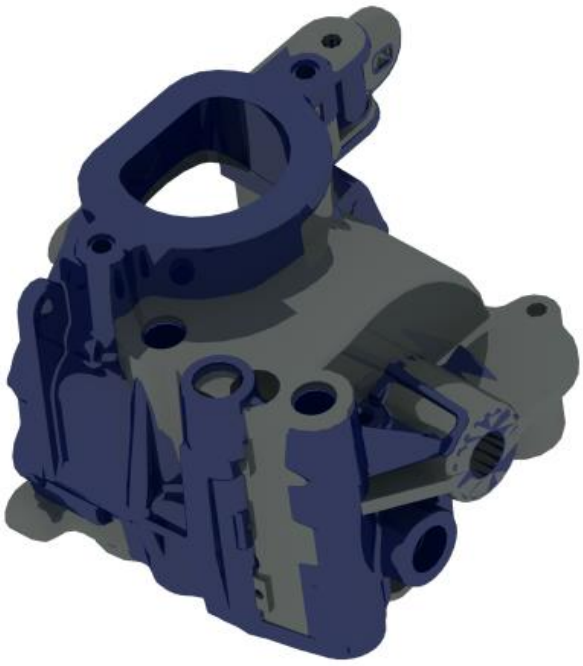


Figure 74. Concept and Commercial Pump Mainbody overlapping.

Figure 75. Concept and Commercial Pump Mainbody overlapping.

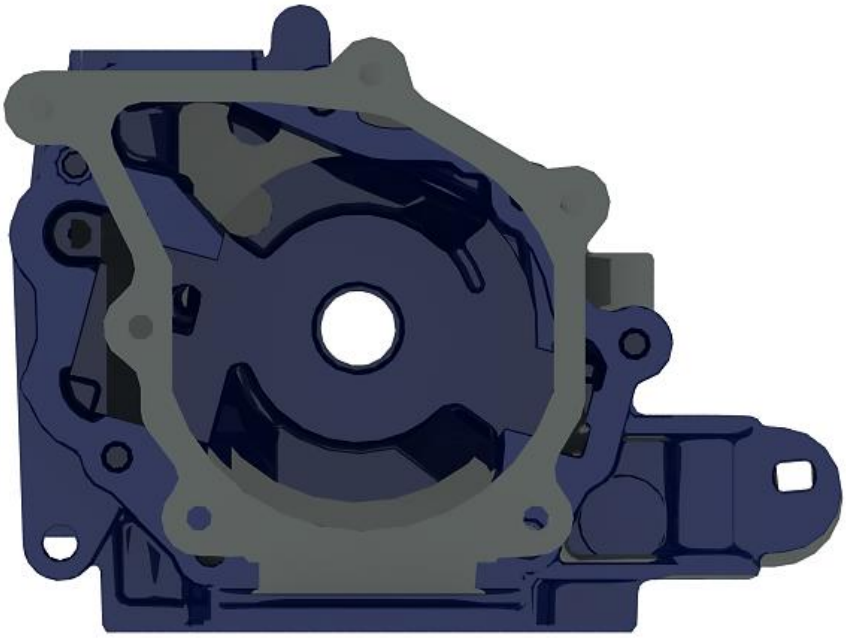


Figure 76.Concept and Commercial Pump Mainbody overlapping (Front View).

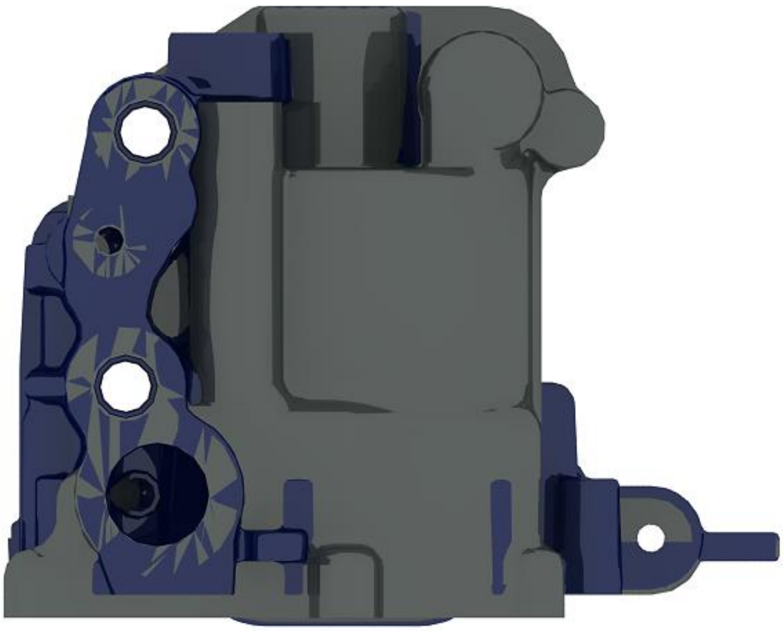


Figure 77. Concept and Commercial Pump Mainbody overlapping (Bottom View).

4.4 Pump Prototype

The images presented are from the non-functional/concept prototype obtained from the concept CAD modulation. The purpose of the prototype is to, however non-functional, allow a better understanding of the system operating principle and parts design this due to the difficulty to explain the same through technical drawings and CAD models.

The prototype was obtained with FDM (Fused deposition Modelling), this technology consists in the controlled deposition of fused thermoplastic. The 3D model (in an STL file) is divided into sections and slices, creating several layers, based in this the extrusion head path is defined. The printer heats the thermoplastics until a semiliquid state and deposits the same in droplets along the defined tool path.

The equipment used was a LeapFrog printer.

4.4.1 Prototype overview

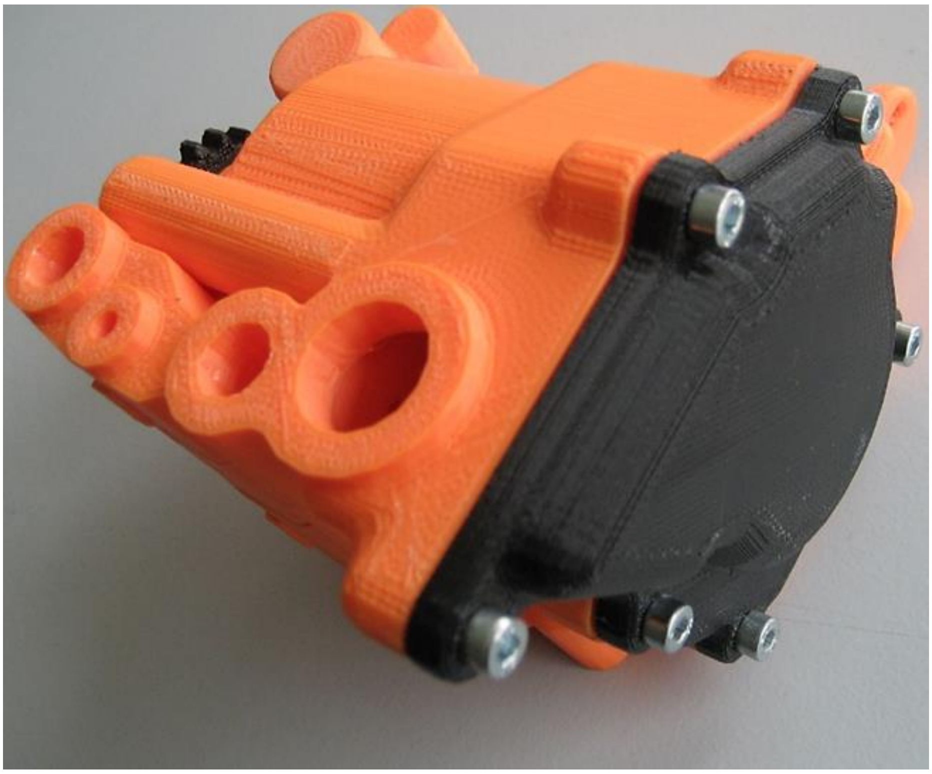


Figure 78. Prototype General View

Figure 79. Prototype General View

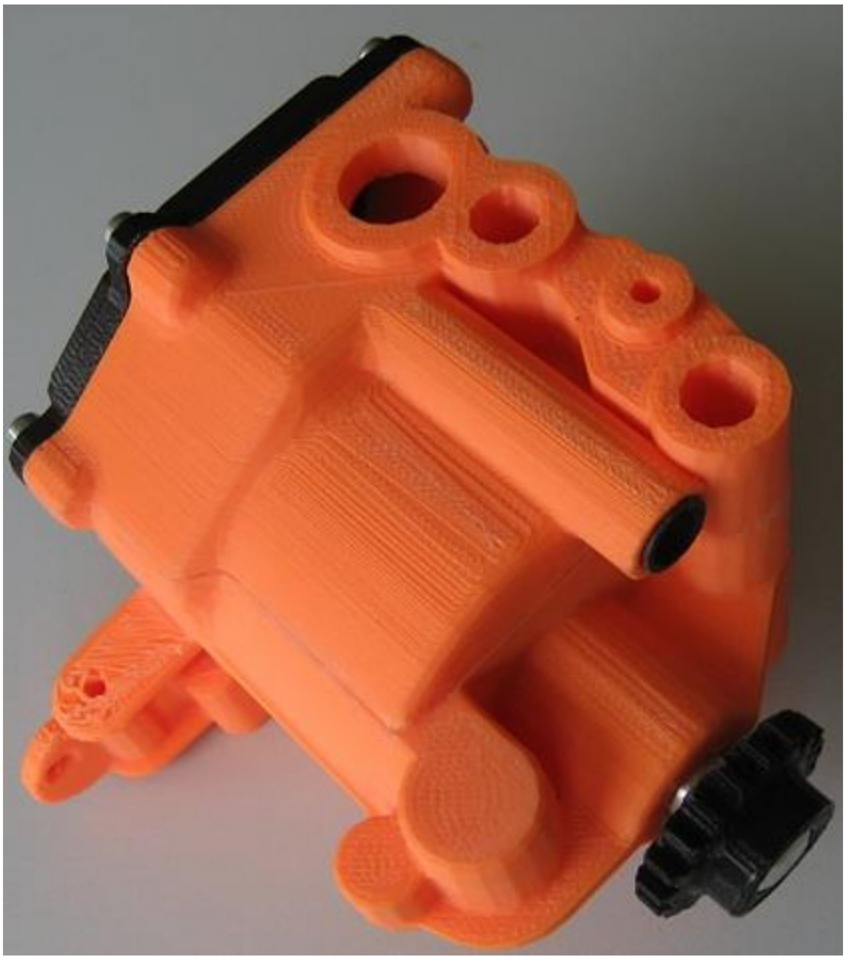


Figure 80. Prototype General View.

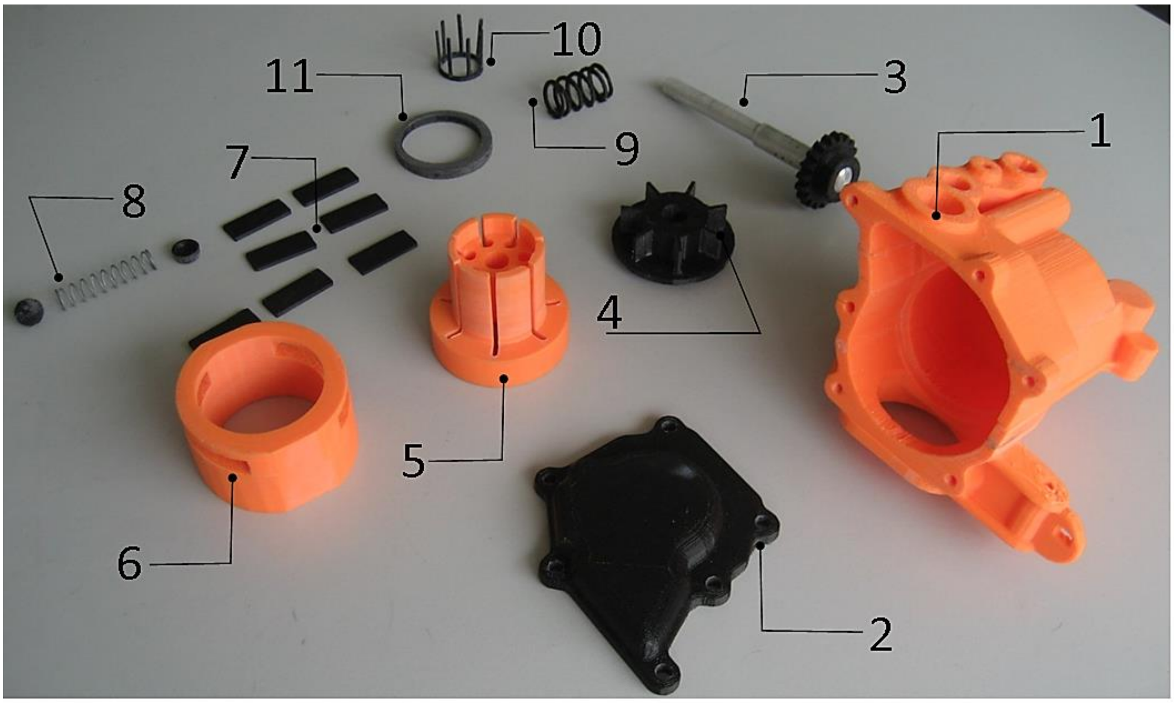


Figure 81. Prototype Parts

The main parts of the prototype are represented in Figure 81.

- 1- Pump Body.
- 2 – Front Cover.
- 3 – Driving shaft with commanding gear.
- 4 – Rotor Sealer.
- 5 – Rotor.
- 6 – Stator (pumping chamber housing).
- 7 – Vanes.
- 8 – Safety discharge valve parts.
- 9 – Control Spring.
- 10 – Vanes holder.
- 11 – Pumping chamber front sealer ring

4.4.2 Prototype Description

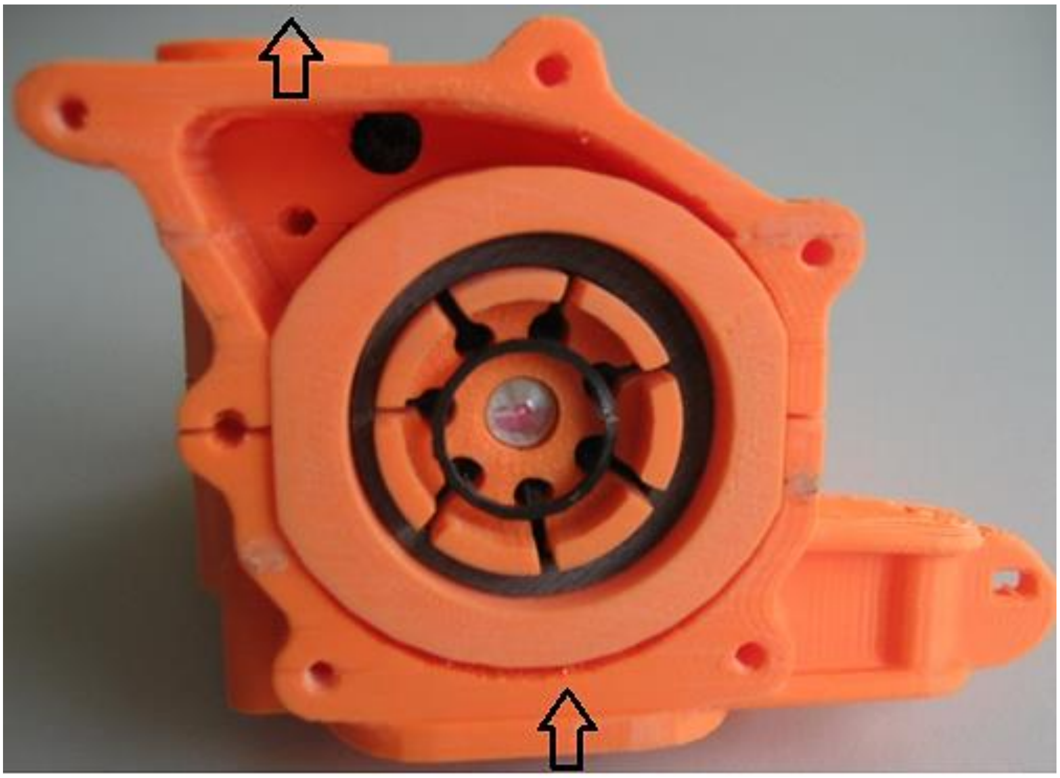


Figure 82. Oil inlet and outlet represented

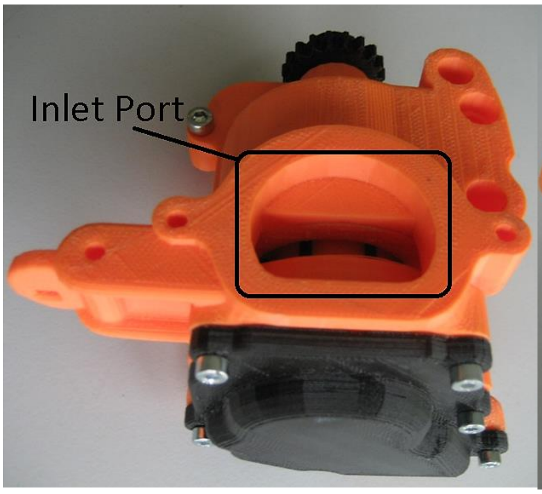


Figure 83. Outlet Port

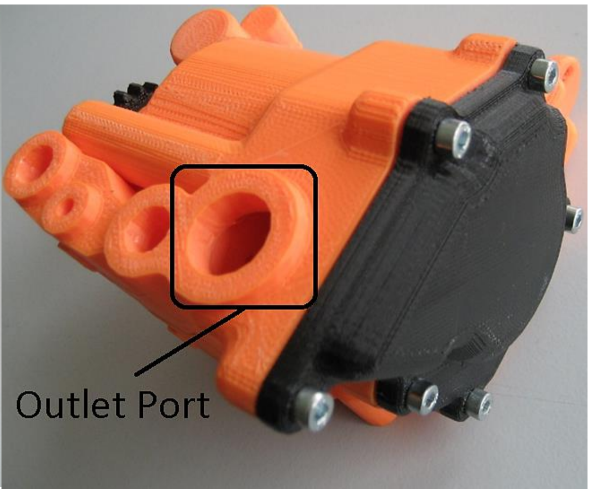


Figure 84. Inlet Port

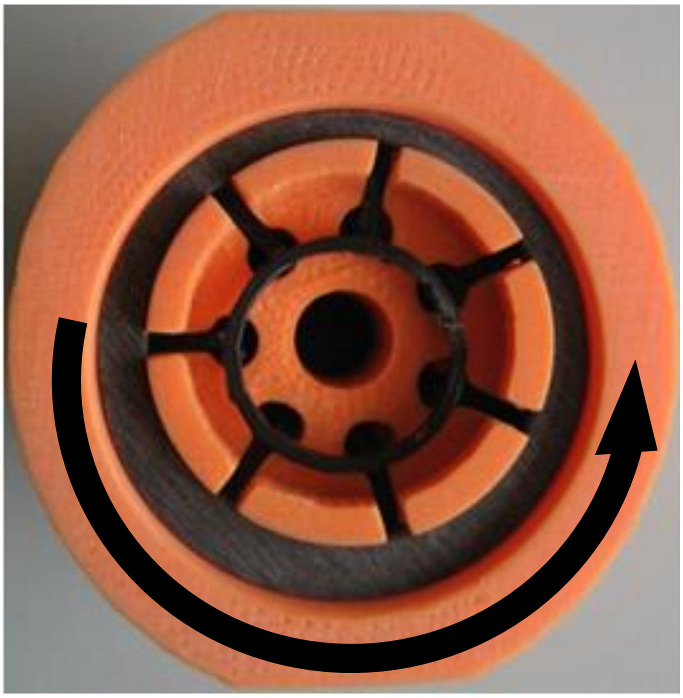


Figure 85. Pumping Set rotation.

The oil is sucked into the pump in the inlet position represented in Figure 82, the rotation of the vanes set, driven by the rotor, lead to the oil compression and expelling, pressure rises due to the diminution of volume in the pumping chamber. This corresponds to a basic operating principle of any vane pump.

The oil is moved from the inlet to the outlet through the pockets created between each two vanes, these pockets are physically created/limited by the Front Sealer Ring, rotor, and pumping chamber housing. Figure 86 represent the rotor set with vanes, front sealer ring, rotor sealer and vanes holder (this element intends to ensure the vanes position when the pump is not operation).

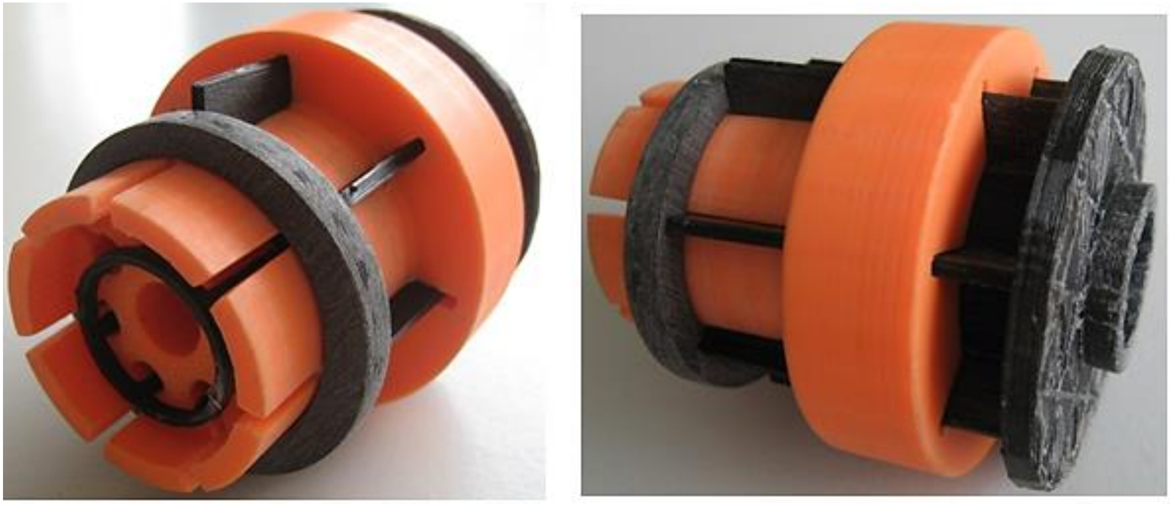


Figure 86. Rotor Set (rotor, vanes, front ring sealer, rotor sealer, vanes holder)

The position of the front sealer and rotor sealer varies due to opposite loads applied on each one, these elements guide the vanes to an outer or inner (inside the rotor enclosure area) position, leading to a higher or lower pumped volume per rotation. Figure 87.

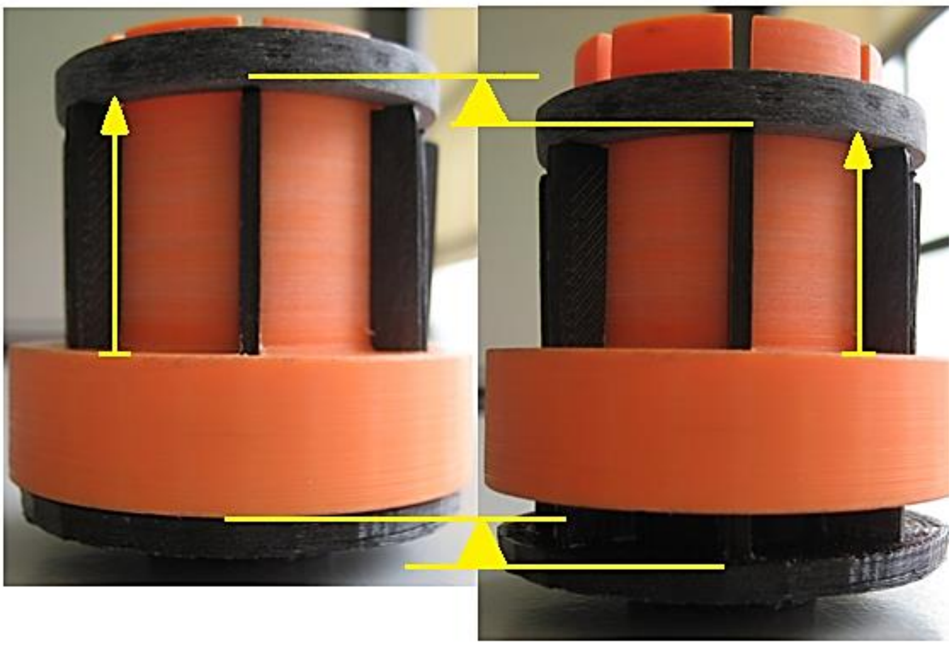


Figure 87. Different displacement positions variation.

The rotor set is enclosure in the pumping chamber which is composed by a ring with two openings at opposite ends, these correspond to the chamber inlet and outlet. Is in the internal walls of the pumping chamber that the vanes are dragged against, ensuring this way the sealing of each pocket.

Figure 88. Pumping chamber and pocket representation.

The pumping rotor set is driven by the pump shaft, the spring is pre-forced in the Rotor Sealer and is supported in the pump body.

Figure 89. Pump pumping rotor set with driving shaft and gear.

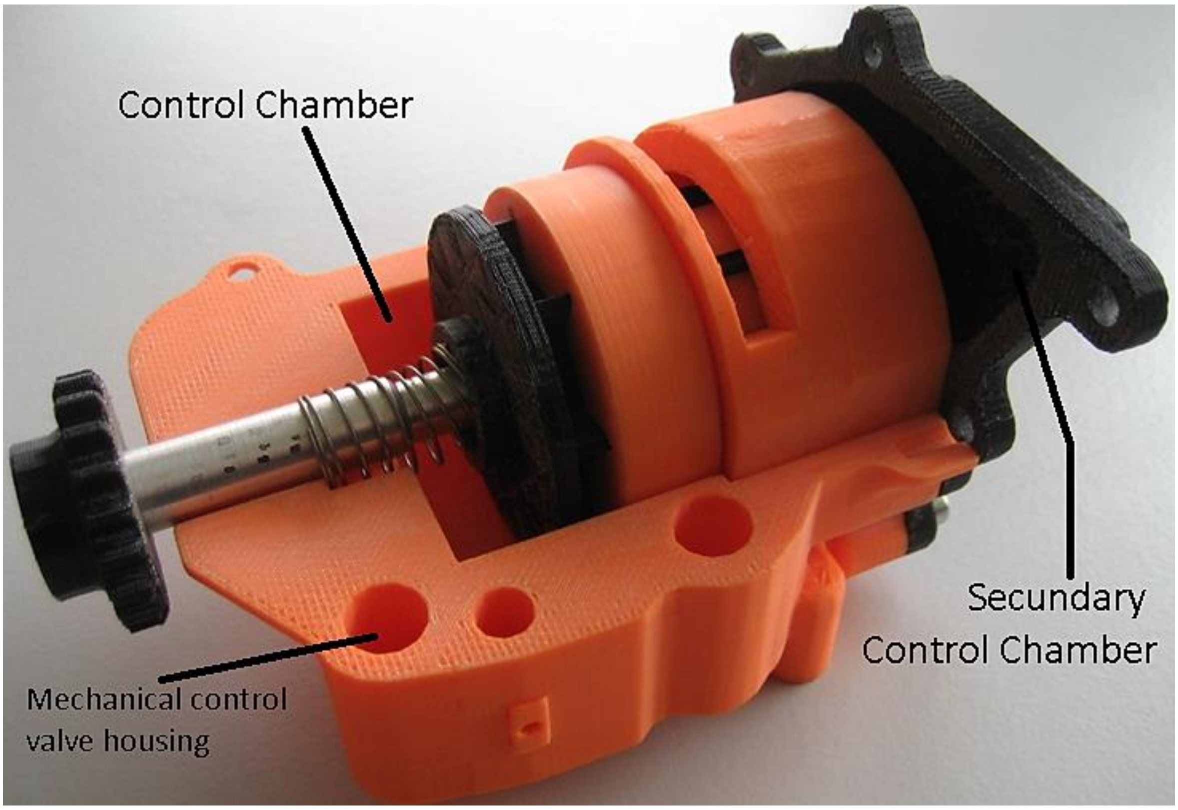


Figure 90. Pump Prototype control chambers

In Figure 90 it's possible to observe the chambers that lead to the control of the moving elements. The balance between the control chamber (which is connected to the mechanical control valve, and assisted by the pre-forced spring) is applied on the Rotor Sealer and the pressure at the Secondary control chamber which is applied in the front sealer ring, define the position of these two elements that consequently define vanes position and displaced volume per rotation.

4.5 Proposed vs achieved

In the process for the product development, several targets were defined in order to have a guidance/objective for the definition of the concept. Some of the defined targets are relative to the effect that the new design will have in the hole vehicle system, this away, taking into account the work developed, it is not possible to do a direct analysis, however it is possible to analyse some achieved values and do an analysis based in the influence that this ones can have in the final targets.

Relatively to the weight of the pump it was indicated that the medium value for the market designs was 2,3[kg], this specification influences several factors, a heavier element represents more power consumption, fuel consumption, CO_2 emissions. The achieved value for the concept is 1.8[kg], this value fulfil the objectives because although it is obvious that it can vary with an necessary optimization of the mechanical parts, it as has a considerable margin that can vary without exceeding the acceptable value for market.

The response to flow variation needs is also a preponderant specification once it influences several objectives as the volumetric efficiency, the reduction of oil changes necessary, as the pump power consumption. The design obtained allows the fulfilling of the range of flow typically required with the variation of the pumping chamber length Table 8 (Page 92). The chamber was defined in order to fulfil the general need of the market (between 10 and 60 [l/min]), taking into account the dimensions of the concept, and as referred in 'Process for pumping chamber dimensioning' the obtained minimum and maximum flow are 10,56 and 65,29 [l/min] which is near to the defined objective.

Also the price of the pump and reliability/robustness are an important factor in the developed concept, these are mainly evaluated based in the number of parts and number of moving parts. In the concept analysed as reference this was basically composed by 36 parts from where eight were standard screws a seven were vanes (all the same component), the moving parts in this product were the pumping chamber housing that slides in order to vary the eccentricity and the rotor with the vanes. In the developed concept exist 32 parts, eight are standard screws and seven vanes (all the same component), this way it's similar to the commercial pump. The number of movable parts is superior with the Front sealer ring, rotor sealer and rotor with vanes. This way the concept is in-line with the commercial designs, the number of parts is similar as the number of movable parts.

Chapter 5

Conclusions and Future Work

5.1 Conclusions

As main objective of this work is the development of an innovative concept for a variable displacement oil pump to be applied in internal combustion engines. This concept was defined to be easily adaptable to the existing systems, meaning that the dimensions and main interfaces need to be kept, this objective was achieved once the concept developed although different from the ones existent respects the same main characteristics. Also in this work its present a revisions of the main solutions for hydraulic pump and more specifically variable displacement oil pumps applied to the automotive market.

The process for development of the concept respected the regular procedure with definition of objectives based in the expectations of the market/client, with base in this it was defined a series of specifications for the pump. A series of the specifications do not depend directly on the concept, but on the effect that this concept as in the global system of a vehicle, this way in the developed work it's not possible to thoroughly evaluate or measure this. However it is possible to evaluate some characteristic as the complexity of the concept, weight, fulfilling of the main requirements in flow delivery, and based in the analysis made, it is possible to affirm that the developed concept its in-line with the target market values.

The main evaluation to be made is in the mechanical concept from the point of view of concept and not final product, in this point of view it is possible to affirm that the present work represents an innovation in the market of variable displacement oil pumps for the automotive area, the operating principle in which the concept is based is inexistent in the current market solution and even in patents referent to pumps for the most diverse applications. The construction of a prototype allowed the validation of the concept as such, besides being a relevant step in the development of a concept, it also allows a better understanding of the pump operating principle.

The concept also presents a level of simplicity of mechanism in line with the ones that existent, allowing the integration of ’standard’ solutions as the vanes operating system, the mechanical control valve and the electro valve into an innovating system, this respecting at the same time the capability of the system to be integrated in existent systems.

It is possible to affirm that this dissertation although does not present a fully validated final product, contributes for the creation of a new technology in the growing market of variable displacement pumps, presenting a new concept fully adaptable to the commercial needs.

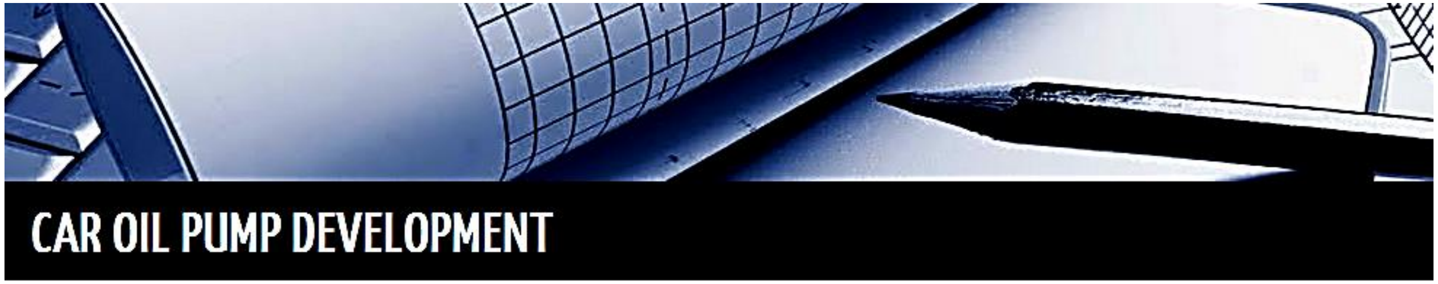
5.2 Future Work

In order to fully evaluate the capability of the developed concept to contribute for the defined needs as fuel and emissions reduction, reduction of the number of oil changes and reduction of the power consumed by the pump, the current project must be validated.

The first step must be the simulation of the pump behaviour both in mechanical solicitation of the components as in efficiency of the fluid behaviour within the system. This simulation allows the dimensional optimization of the defined parts. With the project fully validated it’s possible to, by the creation of a first functional prototype simulate the behaviour of the same in and complete system.

Attachments

Attachment A Questionnaire



Seleccione 3 opções que considere mais importantes no desenvolvimento de uma bomba de óleo de um carro.
Não tem de perceber de carros nem mecânica, basta responder como utilizador/condutor comum.
Obrigado
*Obrigatório

CAR OIL PUMP DEVELOPMENT *

- ☐ Ser 'Environment Friendly'
- ☐ Permitir diminuição do consumo de combustivel
- ☐ Durabilidade / Fiabilidade
- ☐ Custos de manutenção baixos.
- ☐ Fácil de reparar/substituir (em caso de avaria)
- ☐ Preço baixo ou compensador a médio prazo.
- ☐ Permitir redução do número de mudanças de Óleo
- ☐ Melhorar performance geral do carro

[Enviar]

Attachment A. Questionnaire.

Attachment B Quality Function Deployment Matrix

	Weight	General Dimensions	Volumetric Efficiency	CO2 Emmisions Level	Limit Km to oil change	Range of required flow values	Range of Pressure Values	Oil acceptable viscosity	Rigidity	Density	Oil acceptable Temperature Range	Binary/Torque	Power	Fuel Consumption	Limit Price
Weight															
General Dimensions	+														
Volumetric Efficiency															
CO2 Emmisions Level	+		+												
Limit Km to oil change			++												
Range of required flow values			+												
Range of Pressure Values						+									
Oil acceptable viscosity			-			-	--								
Rigidity	-	--													
Density									+						
Oil acceptable Temperature Range							+	+							
Binary/Torque						-									
Power	+			+		-						++			
Fuel Consumption				+								+	++		
Limit Price	-	-	-	-	+						-			+	
Improvement Driver >>>>	↓	↓	↑	↓	↑	O	O	O	↑	↑	O	↓	↓	↓	↓

Likert Scale	1	2	3	4	5
Kano	L	O	E		
Sales Argument	1	1,2	1,5		

Correlation	
++	Positive Strong
+	Positive weak
	Nonexistent
-	Negative weak
- -	Negative Strong

Correlation between customer requirements and product		
⦿	**Strong**	9
○	**Moderate**	3
△	**Weak**	1

Improvement Driver	
O	Value variation without influence
↑	The higher the better
↓	The lower the better

Costumer Requirements		Weight	General Dimensions	Volumetric Efficiency	CO2 Emmisions Level	Limit Km to oil change	Range of required flow values	Range of Pressure Values	Oil acceptable viscosity	Rigidity	Density	Oil acceptable Temperature Range	Binary/Torque	Power	Fuel Consumption	Limit Price	
Envyronment Friendly	Reduce CO2 Emissions				9									3	3	3	
	Reduce Fuel Consumption				1								3	3	9	3	
Maintenance	Reduce number of oil changes					9	3					3				3	
	Easy to repair/replace	1	1														
Performance	Lower required power				3		3	3					3	9			
	Respond to flow rate variation needs			9		9	1		3								
	Weight (to be light)	9	3							9	9						
	Improve General Performance	3		1	3			1					9	9	3		
Reliability	Low mechanical wear of parts							1	3	3	3	3				3	
	Be robust	3	1							9	9					9	
	Importance Level	74,9	16,6	158	192	296	107	61,2	80,1	111	111	80,1	159	279	155	202	2083
	Percentual	4	1	8	9	14	5	3	4	5	5	4	8	13	7	10	100
	Unit	**kg**		**%**	**g(CO2)/km**	**km (x 100)**	**L/min**	**bar**	**cSt**	**GPa**	**g/cm^3**	**ºC**	**Nm @ 6000rpm**	**KW @ 6000rpm**	**L/(100)km**	**Euro**	
Bechmarking Técnico de Produto	**Market value**	**2,3**		**70**	**110**	**20**	**10 a 60**	**<7**	**12.5 -16.3**	**26**	**2,7**	**70 - 120**	**1,75**	**0,6**	**5**	**250**	
	Target Value	**<2,5**		**>80**	**<114**	**>35**	**10 a 60**	**<7**	**12.5 -16.3**	**26**	**2,7**	**70 - 120**	**<1.8**	**<0.65**	**<=5**	**<=250**	
	Techical Difficulty	2	3	4	5	4	4	4	2	2	2	4	5	5	3		

Costumer Requirements	Client	Kano	Grau de importância (geral)	Desirable Product (Bechmarking de Mercado)	Market Value (Bechmarking de Mercado)		Plan	Sales Argument	Argumento de vendas	Absolute weight	Relative weight
Reduce CO2 Emissions	3	E	3	4	4		4	1	2	4,5	12,32877
Reduce Fuel Consumption	4	O	4	5	5		3	1	2	3,6	9,863014
Reduce number of oil changes	5	O	5	5	4		4	1	2	6	16,43836
Easy to repair/replace	2	E	2	3	3		1	0	1	0,8	2,191781
Lower required power	5	O	5	4	4		4	1	1	5	13,69863
Respond to flow rate variation needs	4	O	4	4	5		5	1	1	6	16,43836
Weight (to be light)	2	E	2	4	4		2	1	1	1	2,739726
Improve General Performance	4	O	4	5	4		3	1	2	3,6	9,863014
Low mechanical wear of parts	3	L	3	4	4		5	1	1	3,8	10,27397
Be robust	2	L	2	4	3		3	1	2	2,3	6,164384
										37	100

RP: Product Requirements (Quality Designed / Product Specifications)

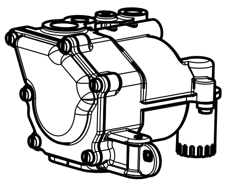
Attachment B. QFD Matrix

Attachment C Assembly Drawings

Qty	PART	MATERIAL
1	Pump Body	Aluminium EN 1706 ENAC-AlSi9Cu3(Fe)
1	Pumping Chamber Body	Steel 42CrMo4
1	Rotor	Steel 42CrMo4
1	Seling Front Ring	-
1	Mechanical valve	Steel 42CrMo4
1	Mechanical valve Controller	Steel 42CrMo4
1	Mechanical Valve Spring	Steel en 10270-3 steel
1	Mechanical Valve Sealer	Steel 42CrMo4
1	Safety discharge valve Sealer	Steel 42CrMo4
1	Saefety Dicharge Valve Sphere	Steel 42CrMo4
1	D12420-ewes (Safety Discharge valve spring)	Steel en 10270-3 steel
1	Electrovalve	Steel (housing material)
1	Rotor Sealer	Steel 42CrMo4
1	Shaft	Steel 42CrMo4
7	vane	Steel 42CrMo4
1	Front Cover	Aluminium EN 1706 ENAC-AlSi9Cu3(Fe)
7	ISO 1207 SCREW M5x14 STEEL GRADE A SLOTTED CHEESE HEAD	Steel Grade A
1	22700-sodemann (Control Chamber Spring	Steel en 10270-3 steel
1	Vanes Holder	Steel 42CrMo4

DESIGNED BY: Joao Batista
DATE: 04-11-2014

Bill Of Materials



SIZE A4

Varialble Displacement Oil Pump

SHEET 1/1

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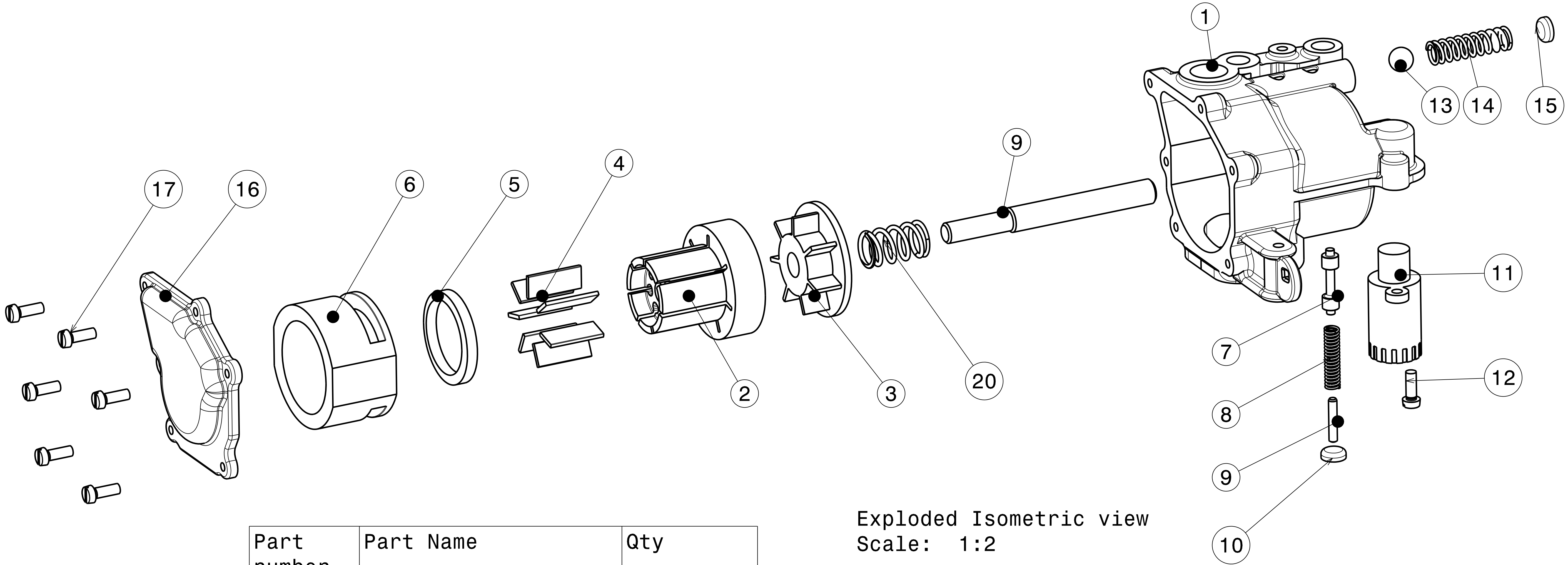
D	_
C	_
B	_
A	_



Exploded Isometric view
Scale: 1:2

Part number	Part Name	Qty
1	PUMP BODY	1
2	ROTOR	1
3	ROTOR SEALER	1
4	VANE	7
5	FRONT SEALER RING	1
6	PUMPING CHAMBER BODY/hOUSING	1
7	MECHANICAL CONTROL VALVE	1
8	VALVE SPRING	1
9	SHAFT	1
10	VALVE SEALER	1
11	ELECTRO-VALVE	1
12	M5X14 SREW	1
13	SAFETY VALVE SPHERE	1
14	SAFETY VALVE SPRING	1
15	SAFETY VAVLE SEALER	1
16	FRONT COVER	1
17	M5X14 SCREW	6

DESIGNED BY: Joao Batista
DATE: 14-10-2014

SIZE A3

Variable Displacement Oil Pump

SCALE

General Exploded View

SHEET 1/1

I	_
H	_
G	_
F	_
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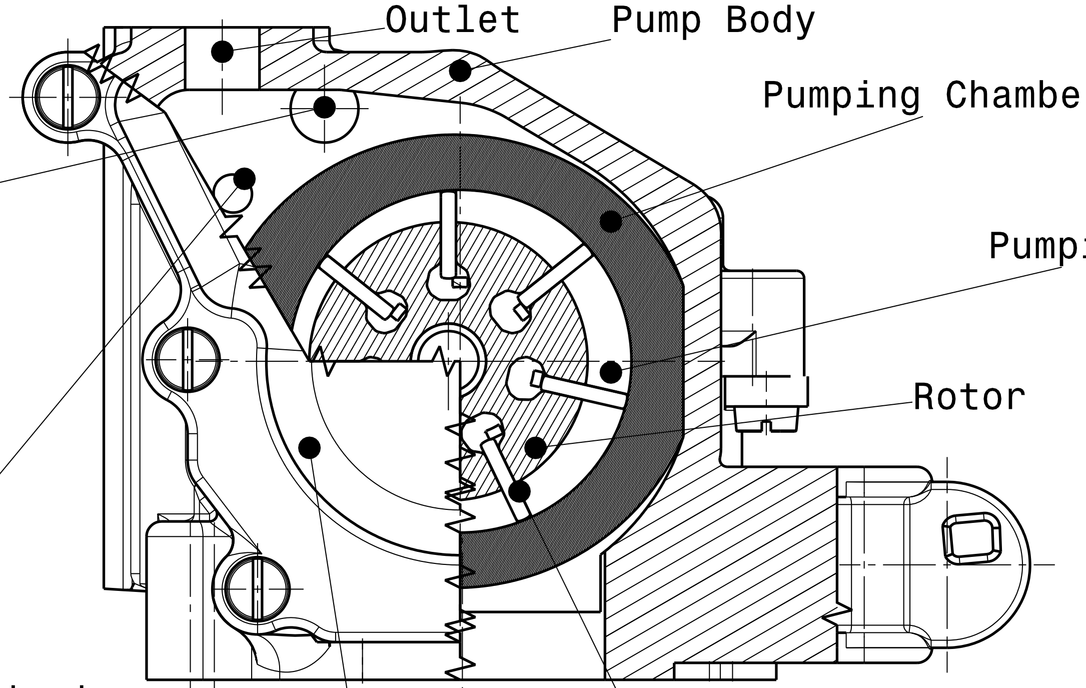
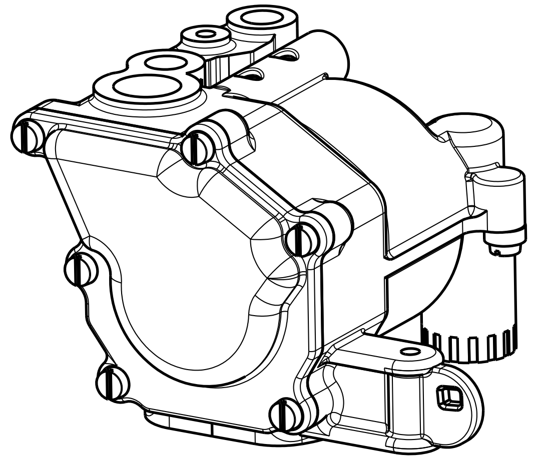
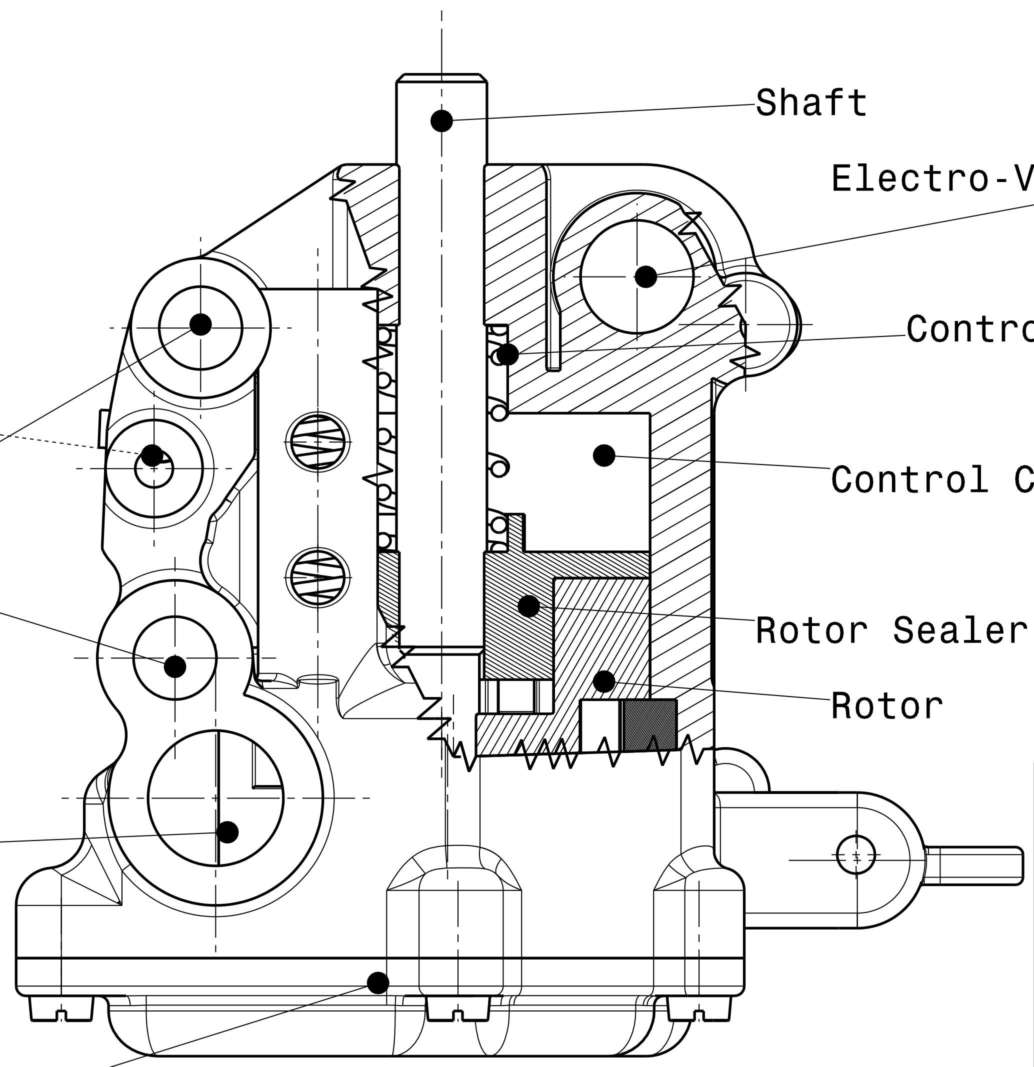
D	_
C	_
B	_
A	_



Front view
Scale: 1:1

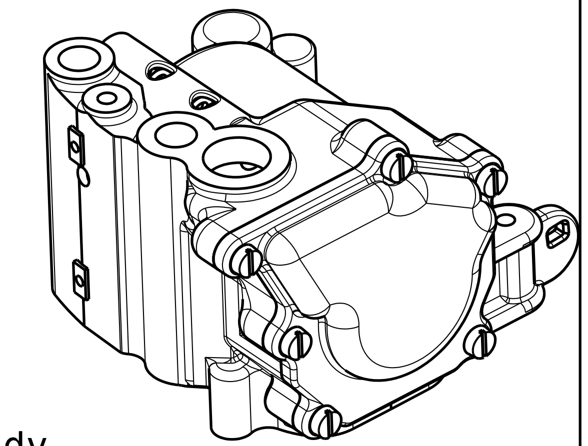


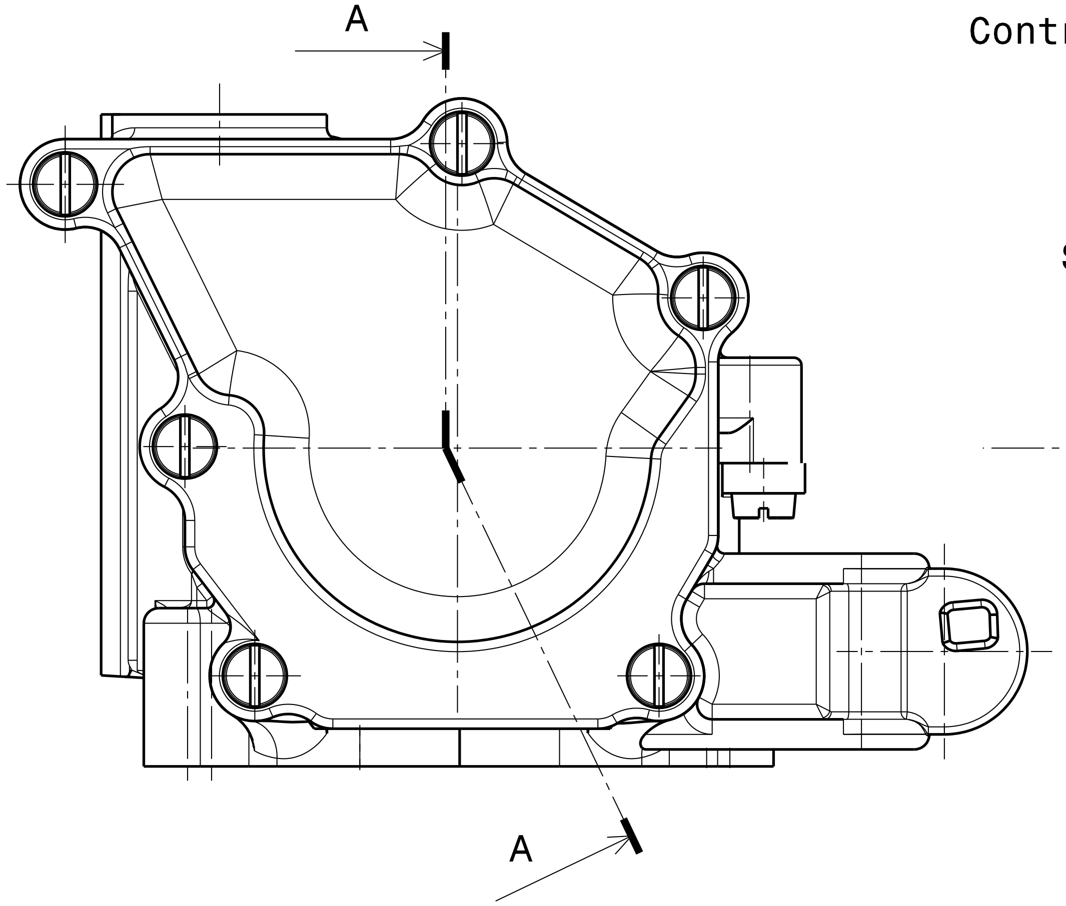


Top view
Scale: 1:1

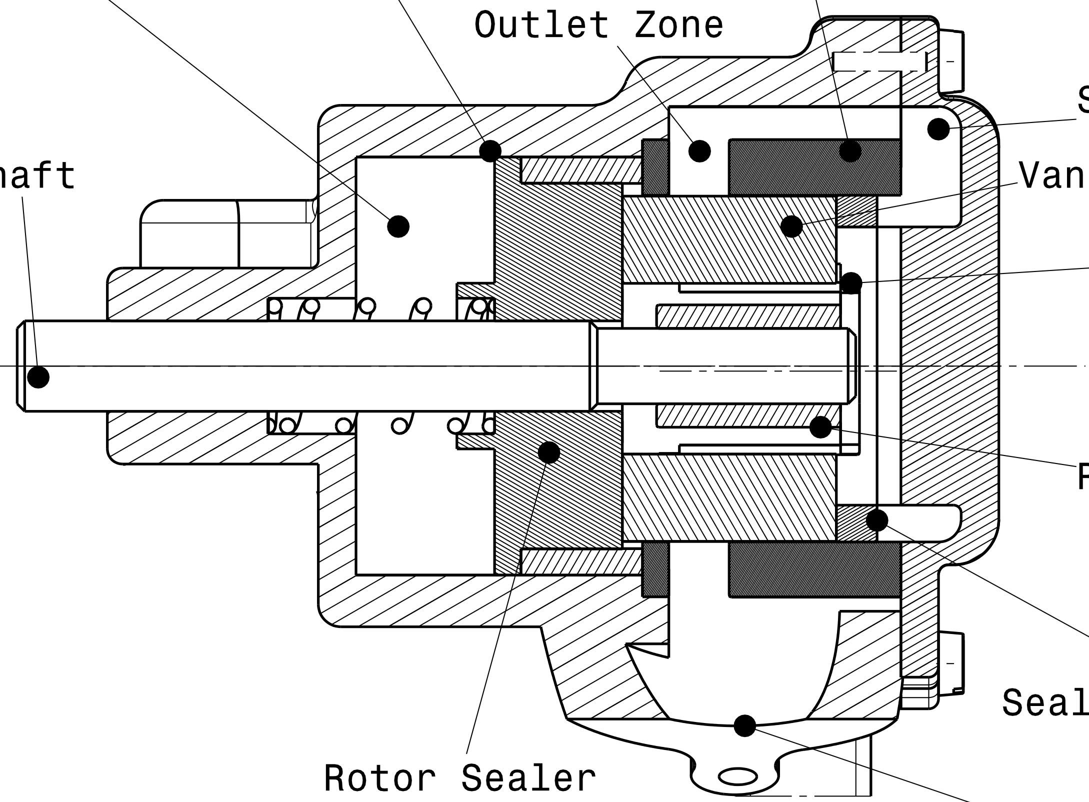
DESIGNED BY: Joao Batista DATE: 14-10-2014			
SIZE A3		Variable Displacement oil Pump	
SCALE 1:1	WEIGHT (kg)	Product Assembly/section View 1	SHEET 1/4

Rev.	
I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_





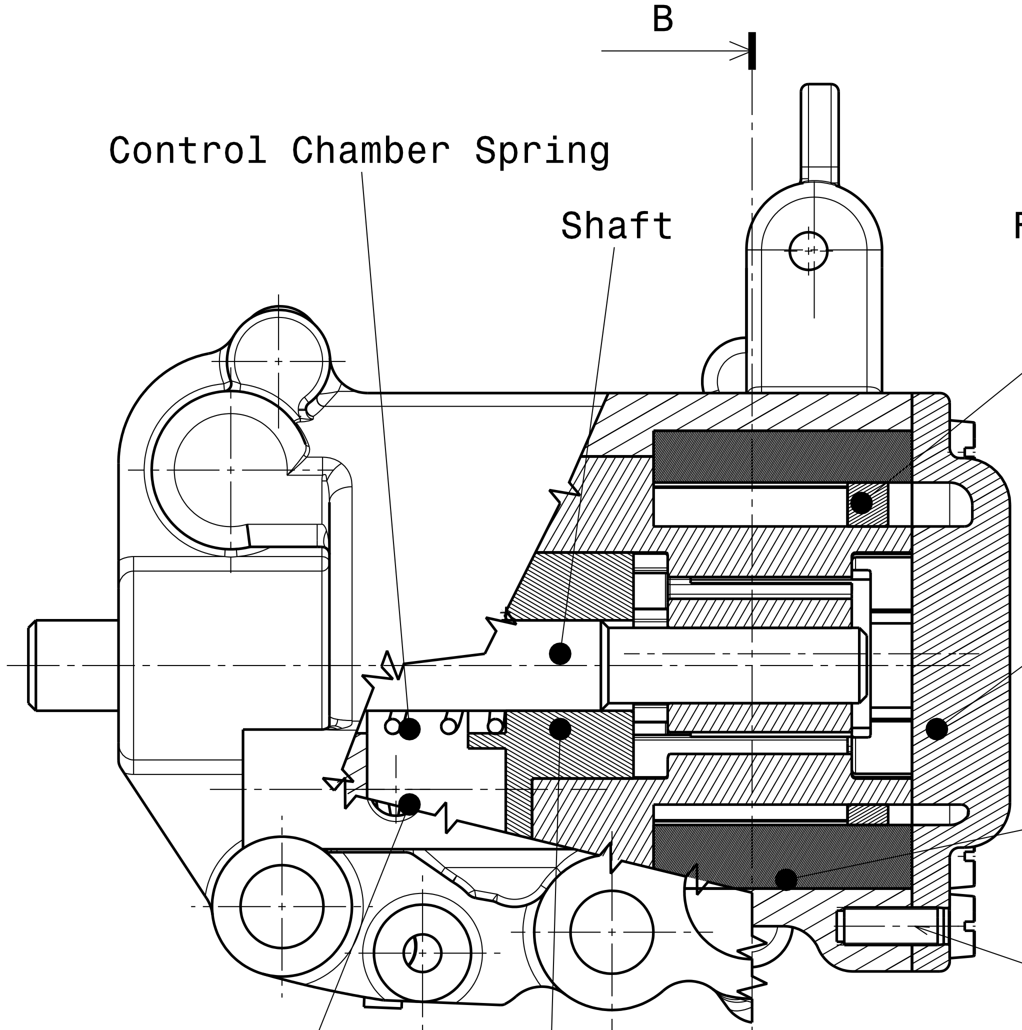
Front view
Scale: 1:1



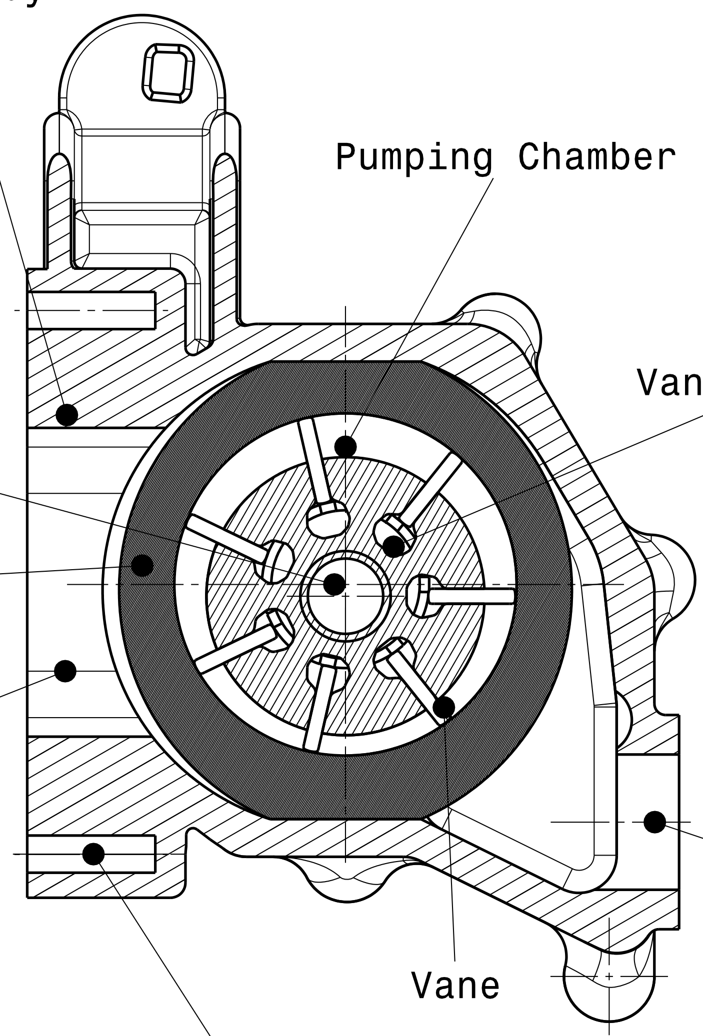
Section view A-A
Scale: 1:1

DESIGNED BY: Joao Batista	
DATE: 14-10-2014	
SIZE A3	Variable Displacement oil Pump
SCALE 1:1	WEIGHT (kg)
Product Assembly/section View 1	SHEET 2/4

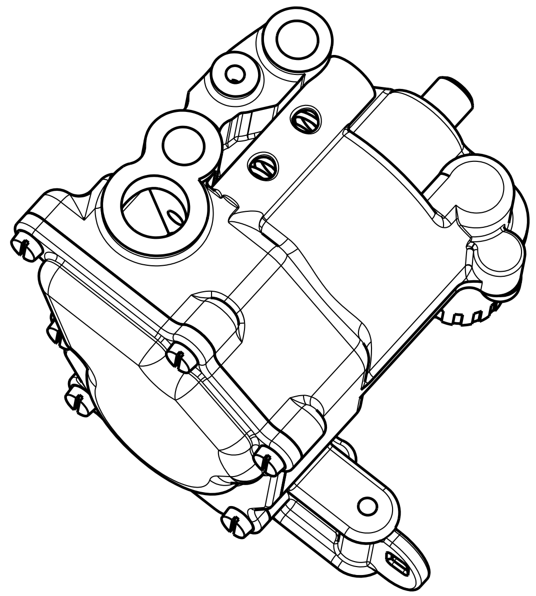
I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_



TopView
Scale: 1:1



Section view B-B
Scale: 1:1



DESIGNED BY: Joao Batista		I	_
DATE: 14-10-2014		H	_
		G	_
		F	_
SIZE A3	Variable Displacement oil Pump	E	_
		D	_
SCALE 1:1 / WEIGHT (kg)	Product Assembly/section View 1 — SHEET 3/4	C	_
		B	_
		A	_

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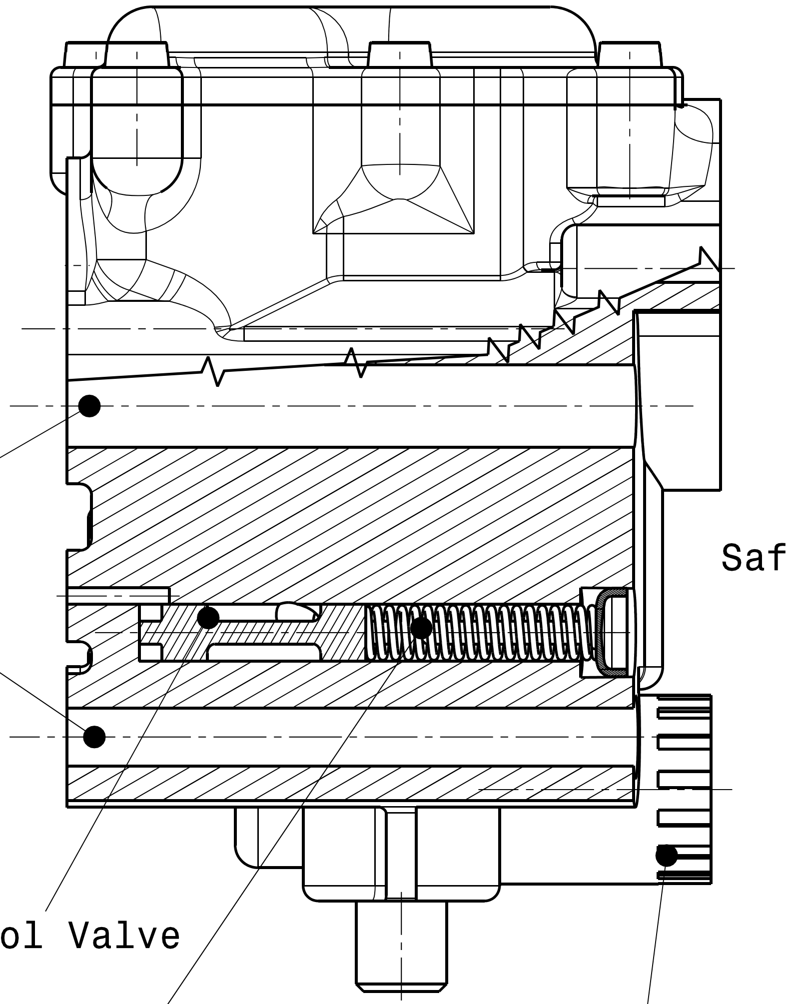
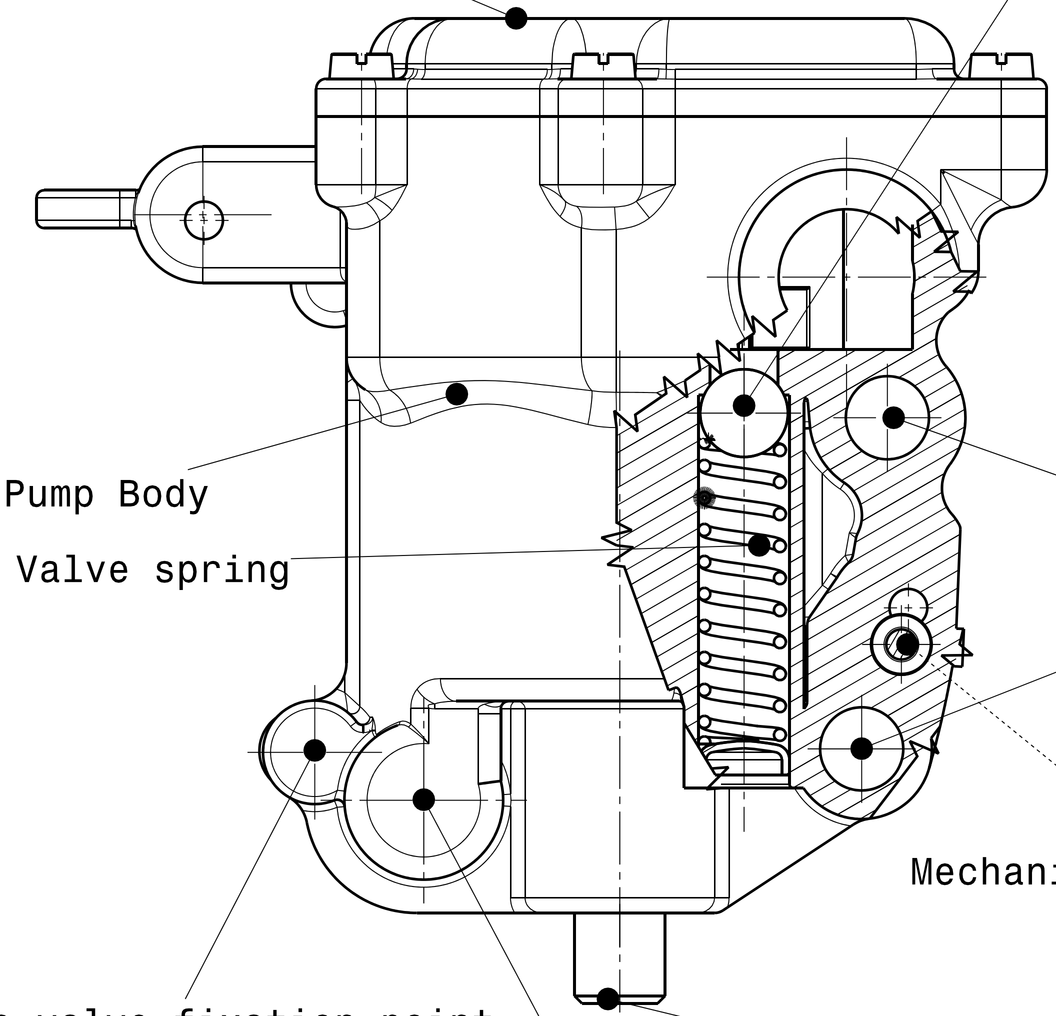


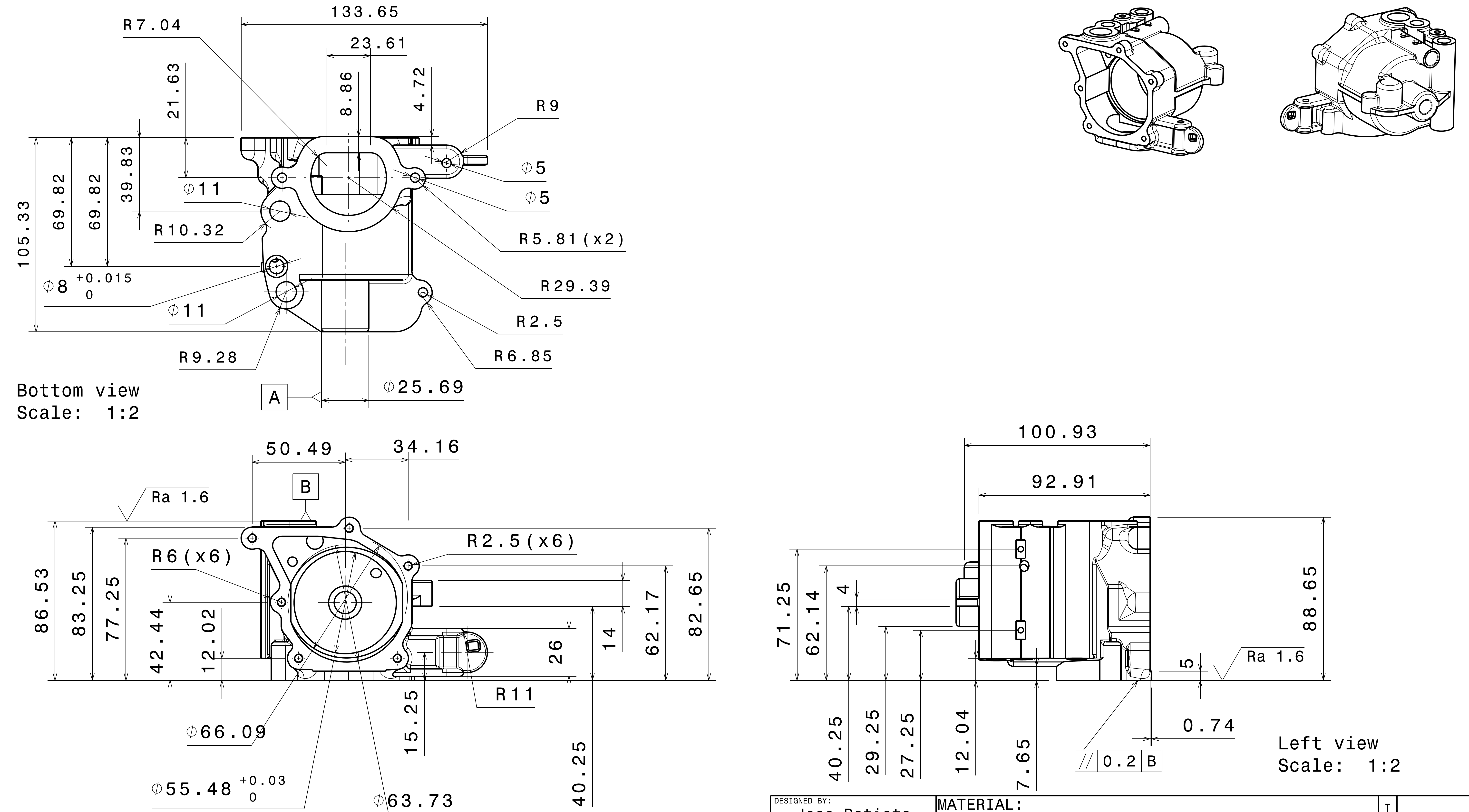
Right view
Scale: 1:1



DESIGNED BY: Joao Batista DATE: 14-10-2014			
SIZE A3		Variable Displacement oil Pump	
SCALE 1:1	WEIGHT (kg)	Product Assembly/section View 1	SHEET 4/4

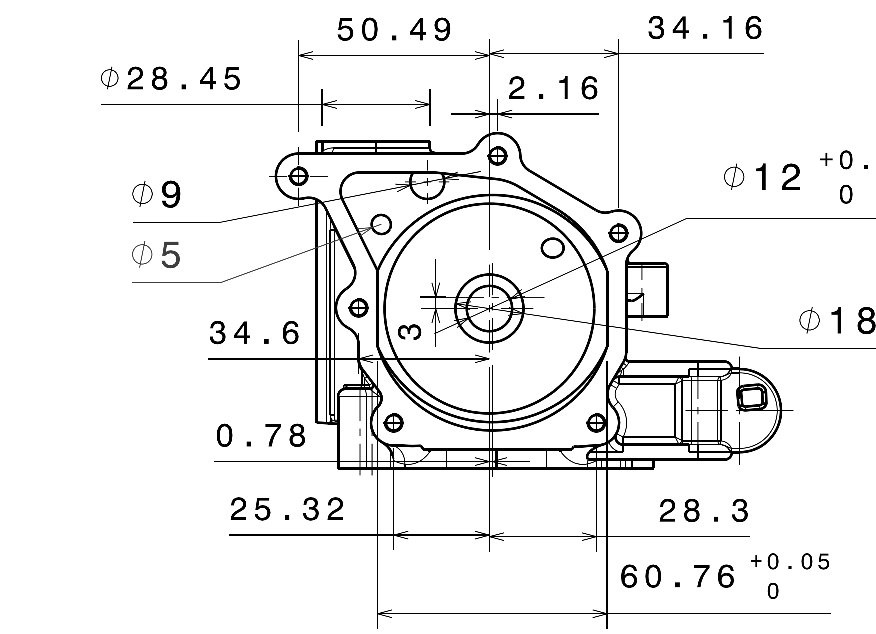
I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_

Attachment D Definition Drawings

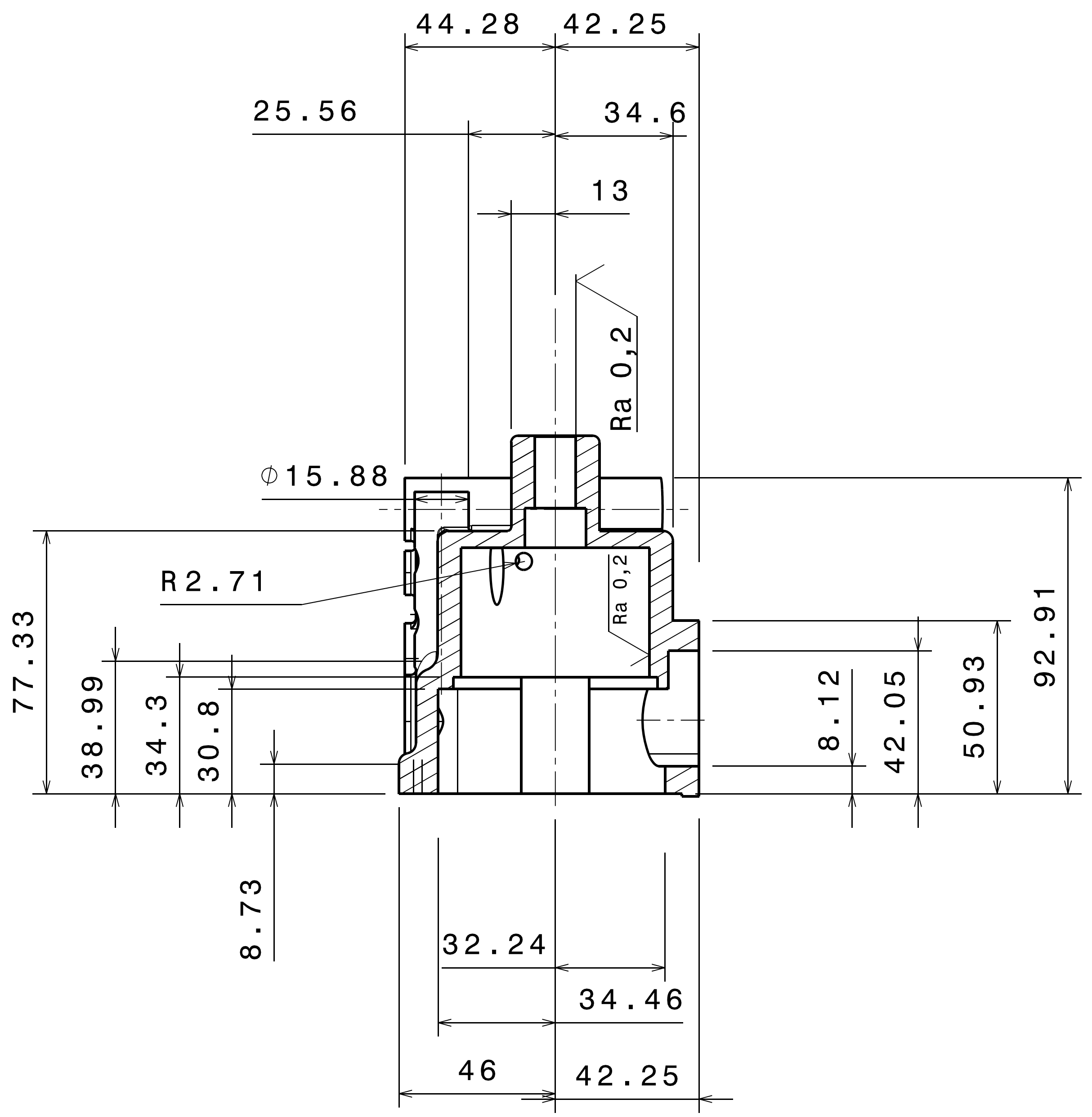


DESIGNED BY: Joao Batista DATE: 27-10-2014	MATERIAL: Aluminium EN 1706 ENAC-AlSi9Cu3(Fe)		
CHECKED BY: DATE:	GENERAL TOLERANCING: ISO 2768 - mk; ISO 8015		
SIZE A3		Variable Displacement Oil Pump	
SCALE 1:1	WEIGHT (kg)	Parr Pump Body	SHEET 1/2

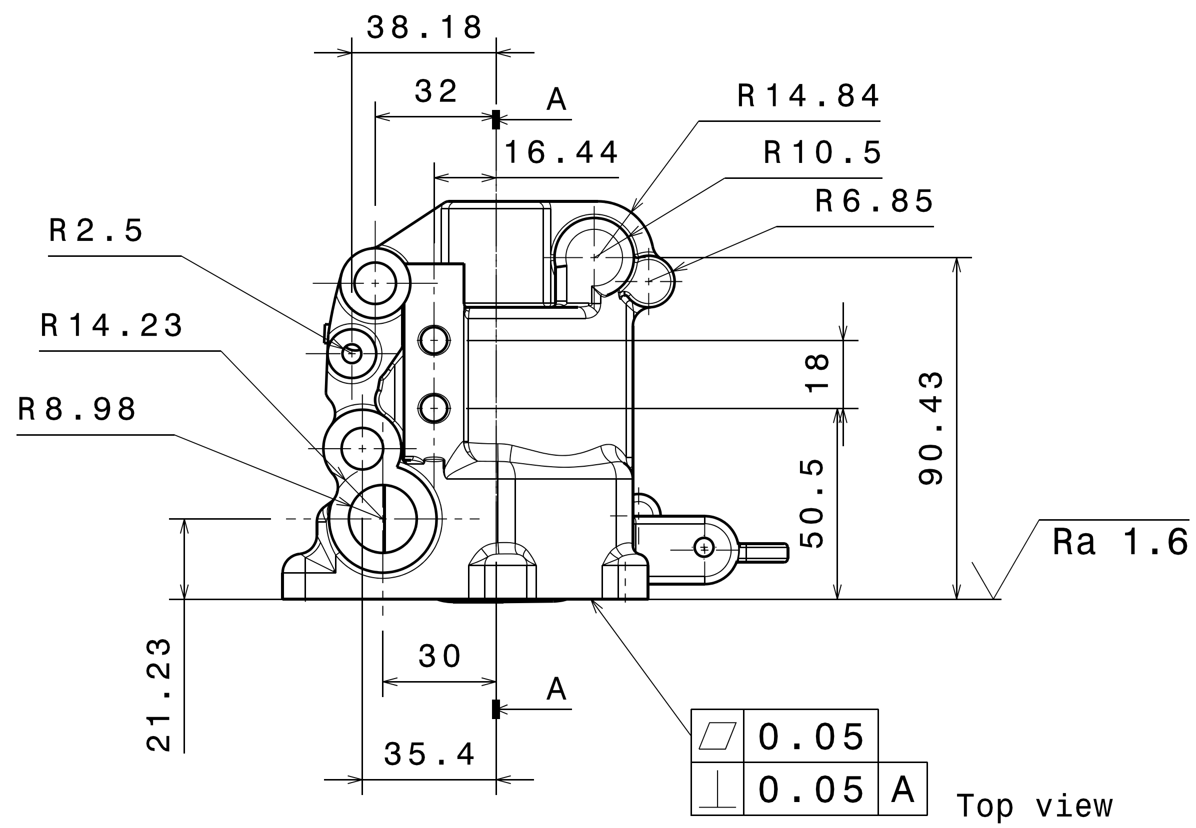
I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_



Front view
Scale: 1:2



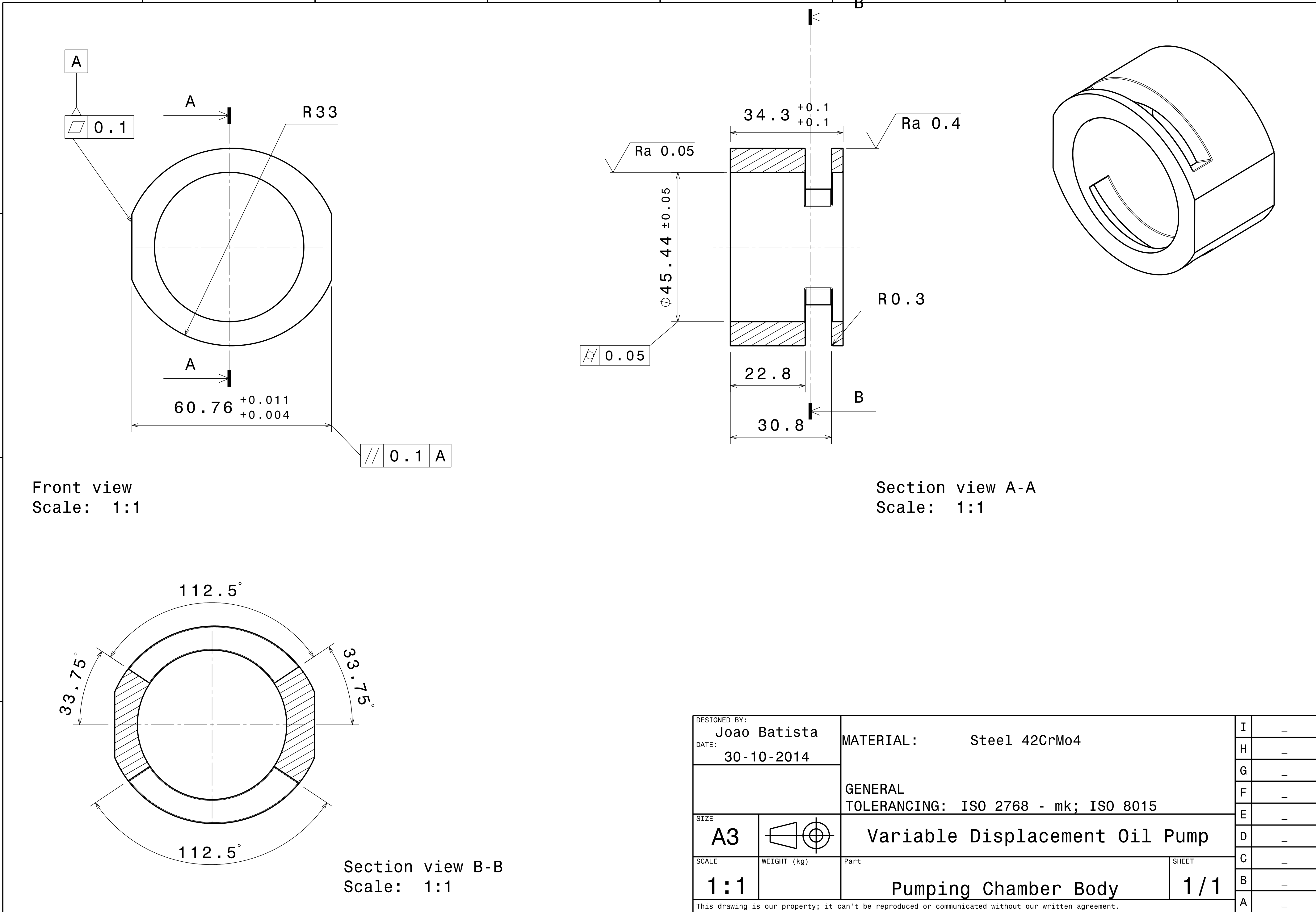
Section view A-A
Scale: 1:2



Top view
Scale: 1:2

DESIGNED BY: Joao Batista DATE: 27-10-2014	MATERIAL: Aluminium EN 1706 ENAC-AlSi9Cu3(Fe) GENERAL TOLERANCING: ISO 2768 - mk; ISO 8015		
SIZE A3		Variable Displacement Oil Pump	
SCALE 1:1	WEIGHT (kg)	Part Pump Body	SHEET 2/2

I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_


A
0.1
A
R33
A
60.76 +0.011 +0.004
0.1 A
Front view
Scale: 1:1
B
34.3 +0.1 +0.1
Ra 0.4
Ra 0.05
Ø45.44 ±0.05
R0.3
0.05
22.8
B
30.8
Section view A-A
Scale: 1:1
112.5°
33.75°
33.75°
112.5°
Section view B-B
Scale: 1:1
DESIGNED BY:
Joao Batista
DATE:
30-10-2014
MATERIAL: Steel 42CrMo4
GENERAL
TOLERANCING: ISO 2768 - mk; ISO 8015
SIZE
A3
Variable Displacement Oil Pump
SCALE
1:1
WEIGHT (kg)
Part
Pumping Chamber Body
SHEET
1/1
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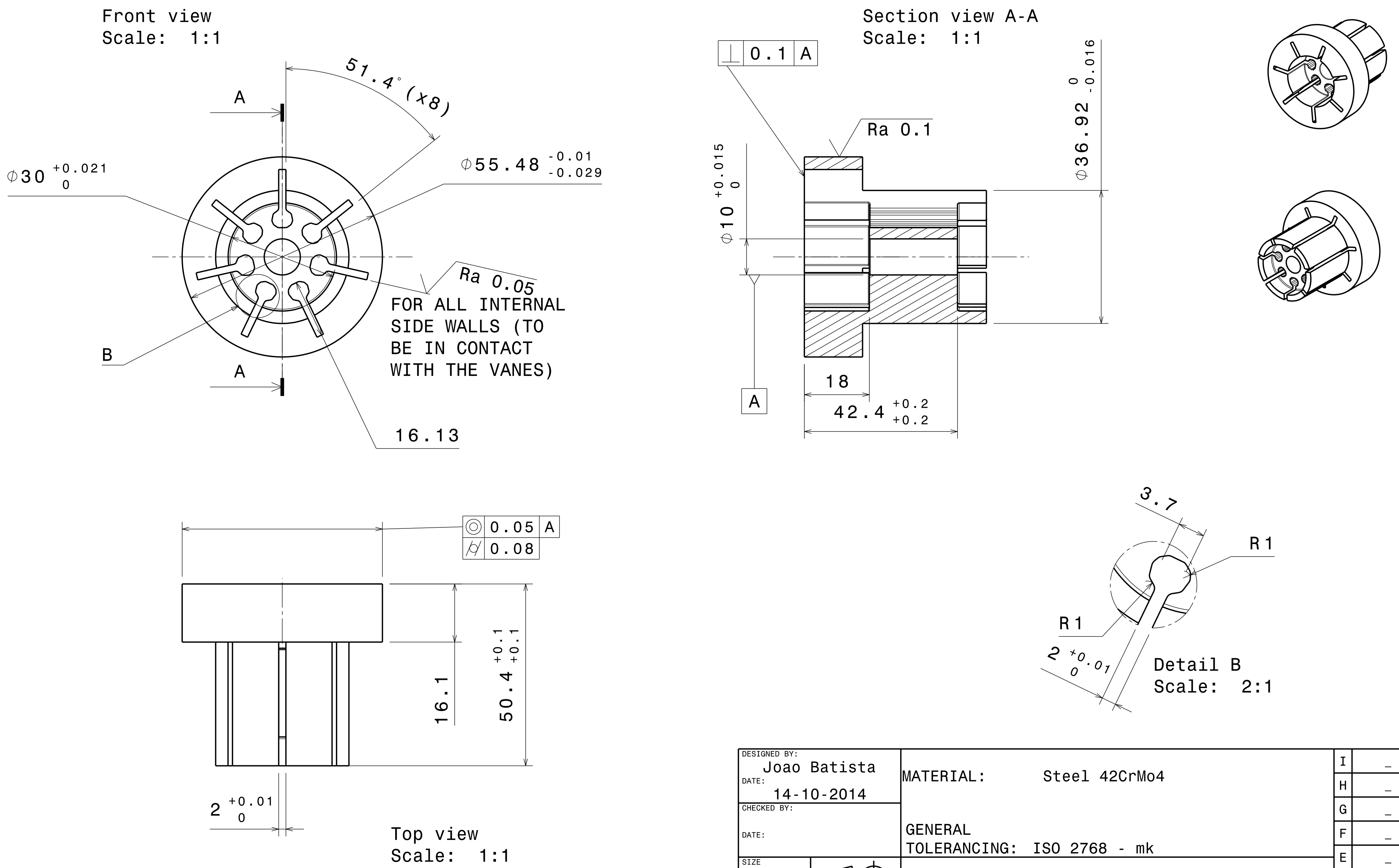


DESIGNED BY: Joao Batista
DATE: 14-10-2014
CHECKED BY:
DATE:

MATERIAL: Steel 42CrMo4

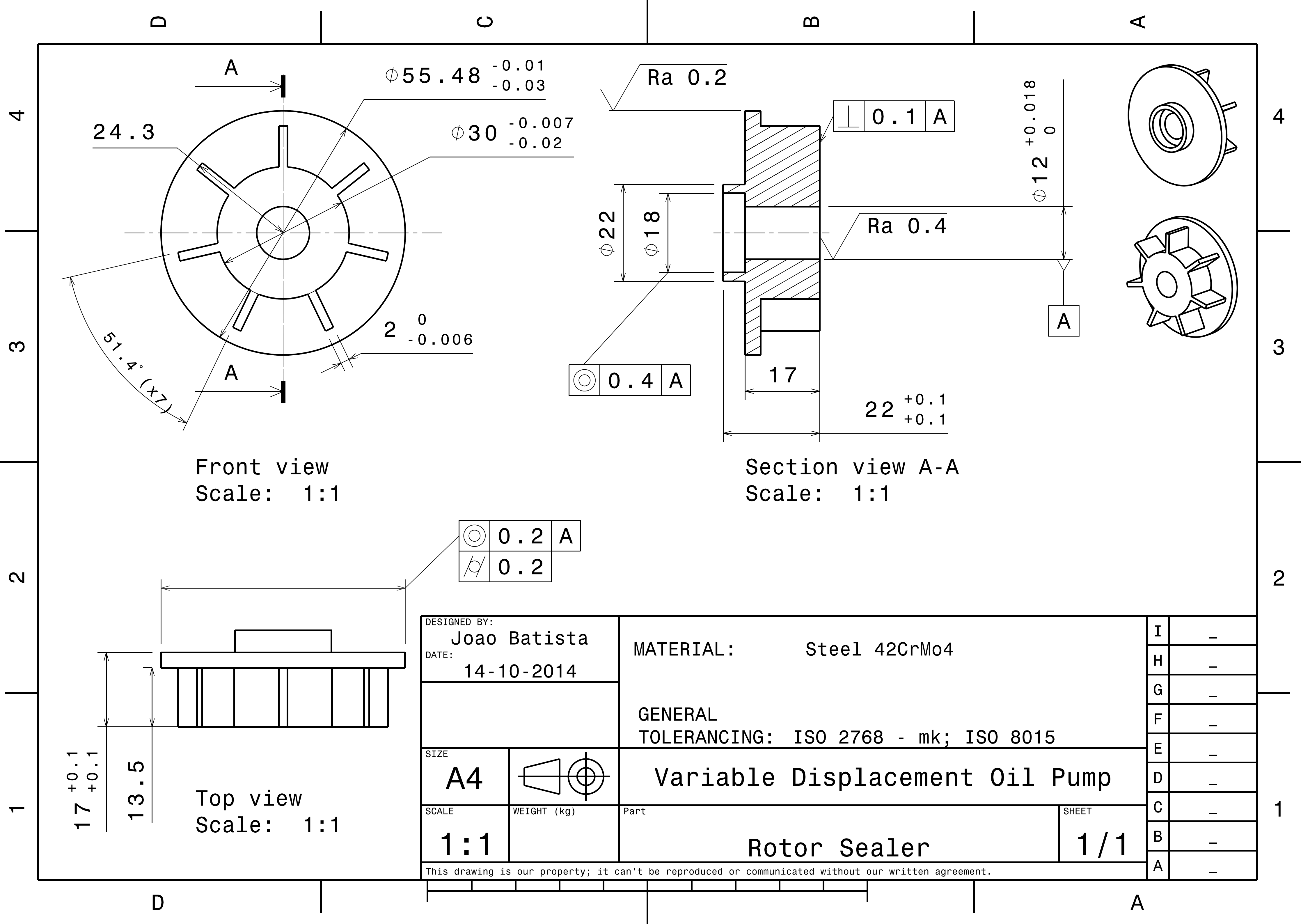
GENERAL TOLERANCING: ISO 2768 - mk

SIZE: A3

Variable Displacement Oil Pump

SCALE: 1:1
WEIGHT (kg)
Part: Rotor
SHEET: 1/1

I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_


A
ϕ55.48 -0.01 -0.03
ϕ30 -0.007 -0.02
24.3
51.4° (x7)
2 0 -0.006
A
Front view
Scale: 1:1
Ra 0.2
⊥ 0.1 A
ϕ22
ϕ18
◎ 0.4 A
ϕ12 +0.018 0
Ra 0.4
A
17
22 +0.1 +0.1
Section view A-A
Scale: 1:1
◎ 0.2 A
⌭ 0.2
17 +0.1 +0.1
13.5
Top view
Scale: 1:1
DESIGNED BY:
Joao Batista
DATE:
14-10-2014
MATERIAL: Steel 42CrMo4
GENERAL
TOLERANCING: ISO 2768 - mk; ISO 8015
SIZE
A4
Variable Displacement Oil Pump
SCALE
1:1
WEIGHT (kg)
Part
Rotor Sealer
SHEET
1/1
I _
H _
G _
F _
E _
D _
C _
B _
A _
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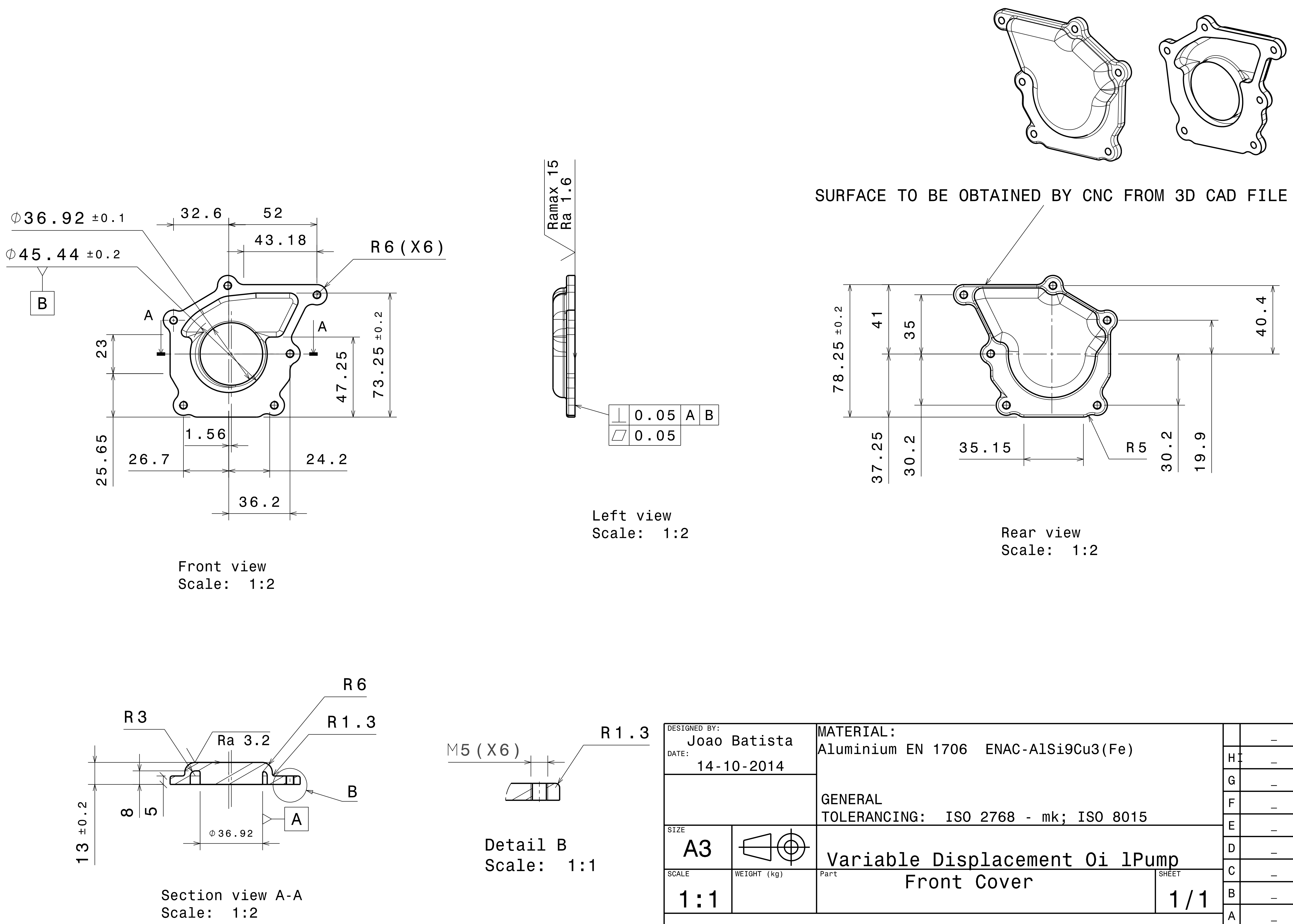


DESIGNED BY: Joao Batista
DATE: 14-10-2014

MATERIAL:
Aluminium EN 1706 ENAC-AlSi9Cu3(Fe)

GENERAL
TOLERANCING: ISO 2768 - mk; ISO 8015

SIZE A3

Variable Displacement Oi lPump

SCALE 1:1

WEIGHT (kg)

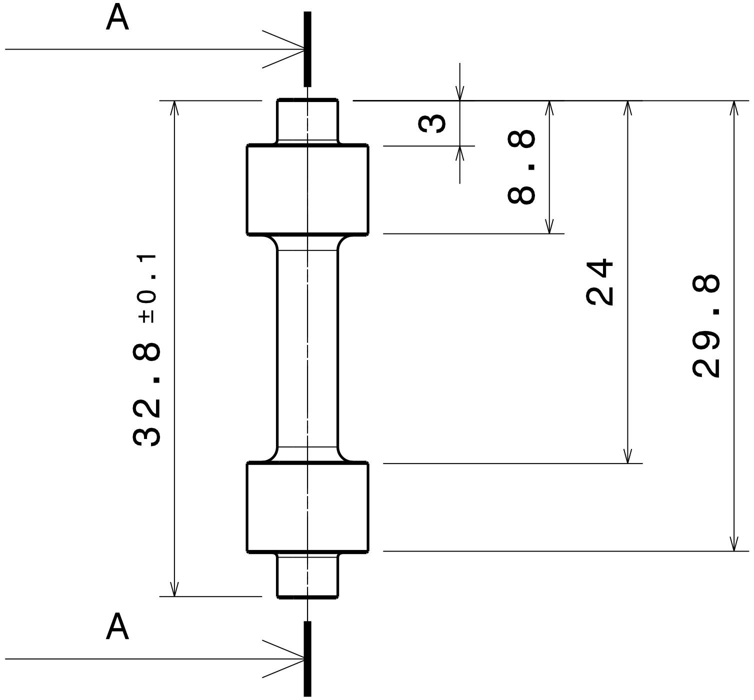
Part Front Cover

SHEET 1/1

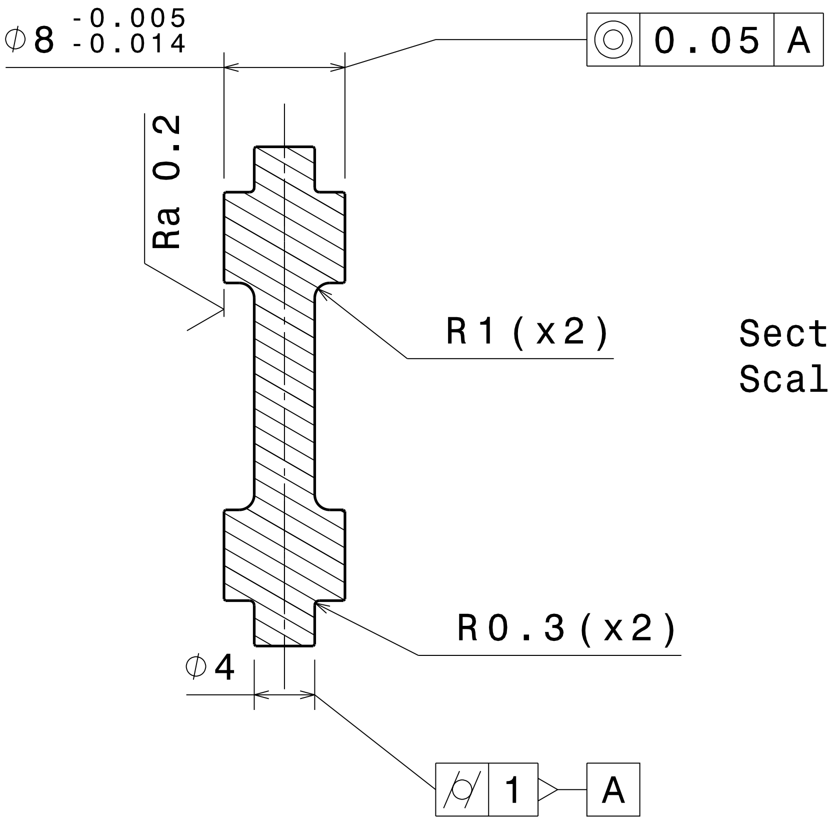
HI	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_

D C B A

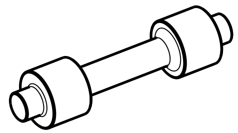
4 3 2 1

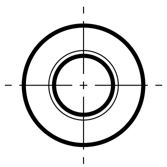


Front view
Scale: 2:1



Section view A-A
Scale: 2:1

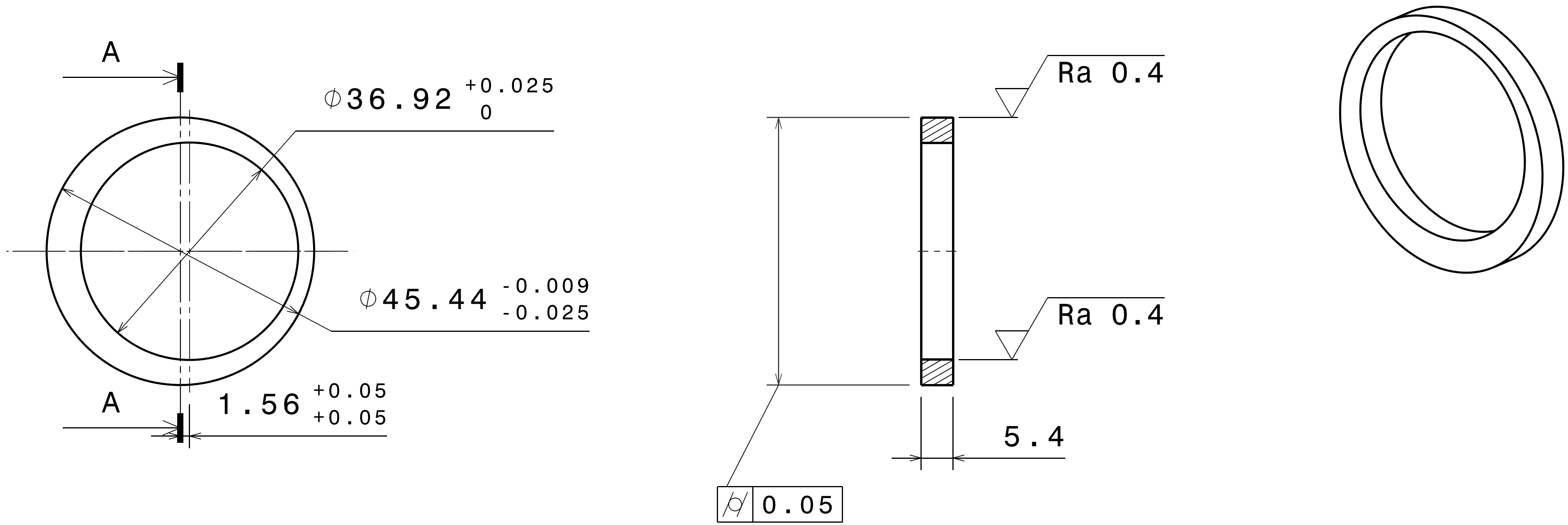




Top view
Scale: 2:1

DESIGNED BY: Joao Batista	MATERIAL: Steel 42CrMo4
DATE: 14-10-2014	GENERAL TOLERANCING: ISO 2768 - mk; ISO8015
SIZE A4	Variable Displacement Oil Pump
SCALE 2:1 / WEIGHT (kg)	Part: Mechanical Control Valve / SHEET 1/1

Rev.	
I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_

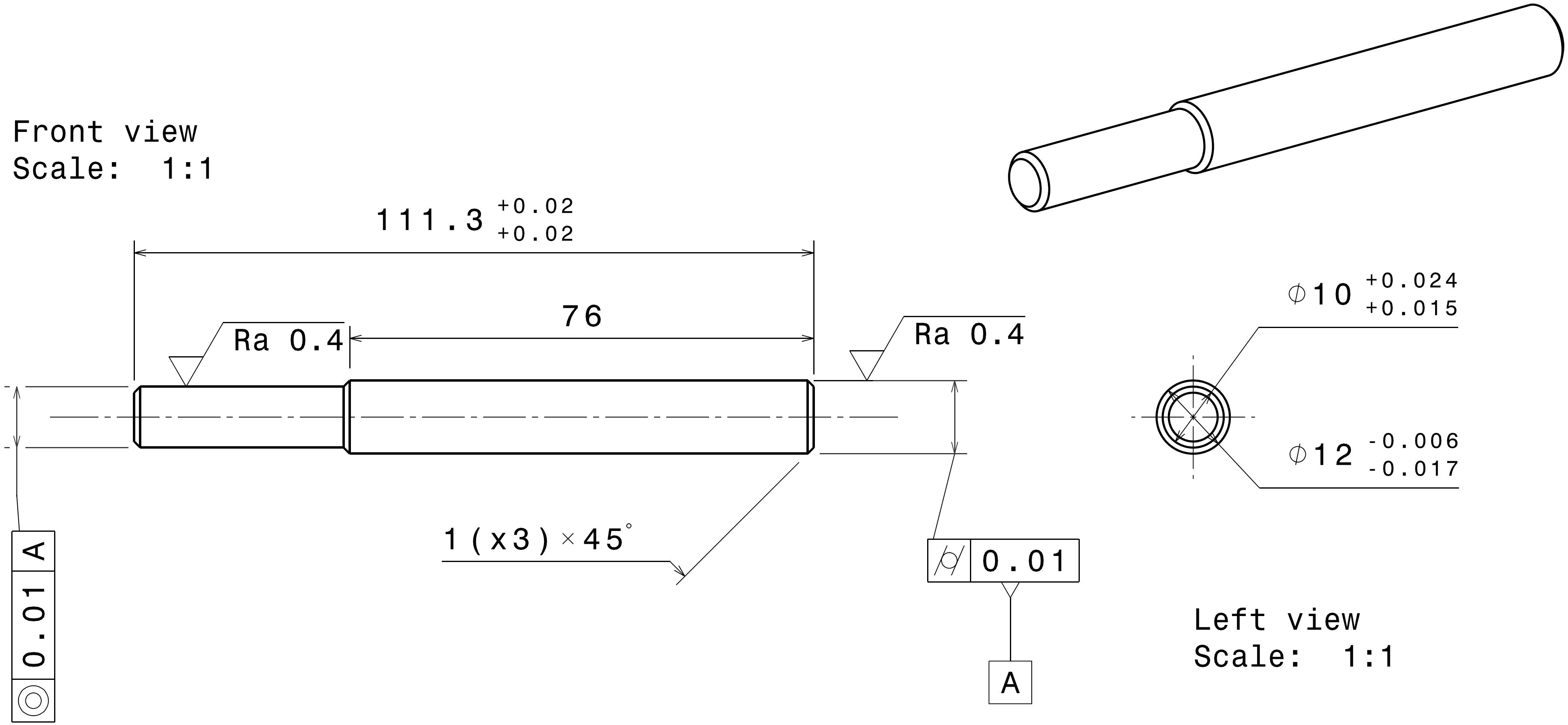


Front view
Scale: 1:1

Section view A-A
Scale: 1:1

DESIGNED BY: Joao Batista DATE: 14-10-2014	MATERIAL: Steel 42CrMo4 GENERAL TOLERANCING: ISO 2768 - mk; ISO 8015		
SIZE A4		Variable Displacement Oil Pump	
SCALE 1:1	WEIGHT (kg)	Part: Front Sealing Ring	SHEET 1/1

I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_



DESIGNED BY: Joao Batista
DATE: 14-10-2014

MATERIAL: Steel 42CrMo4

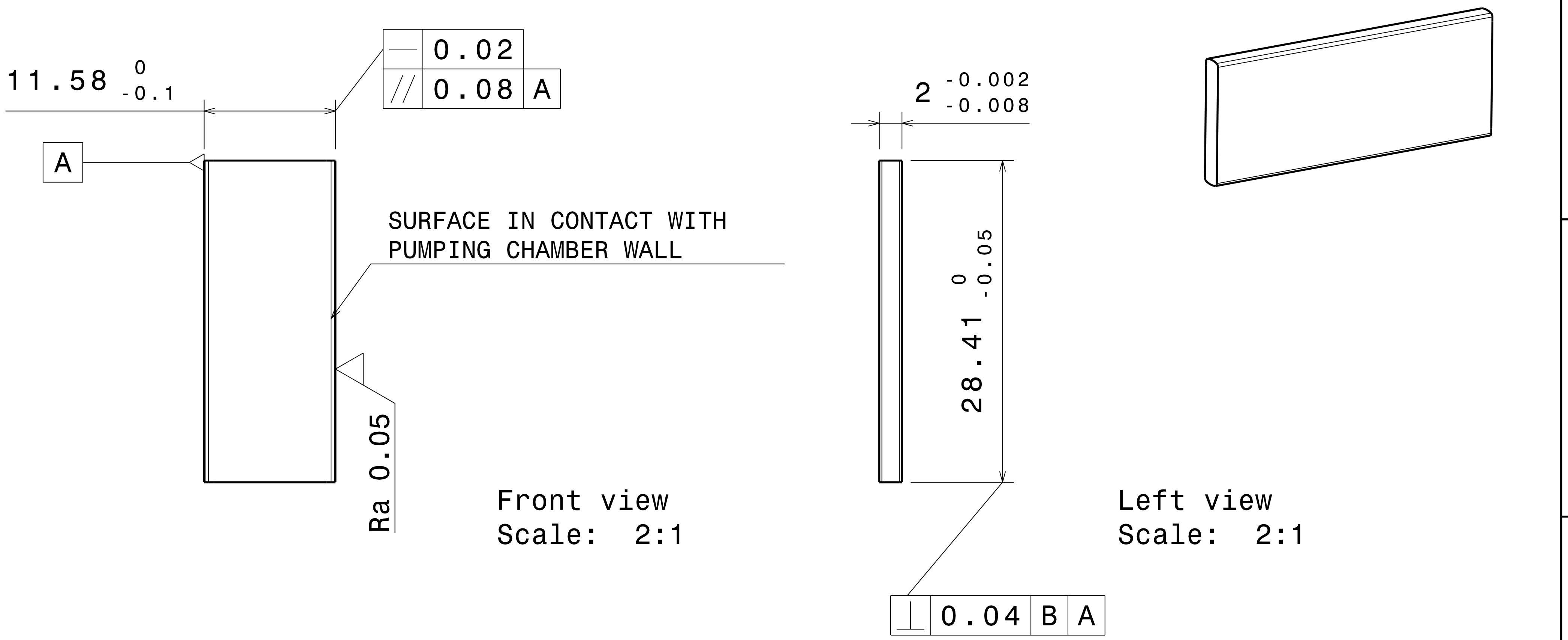
GENERAL TOLERANCING: ISO 2768 - mk; ISO 8015

SIZE: A4

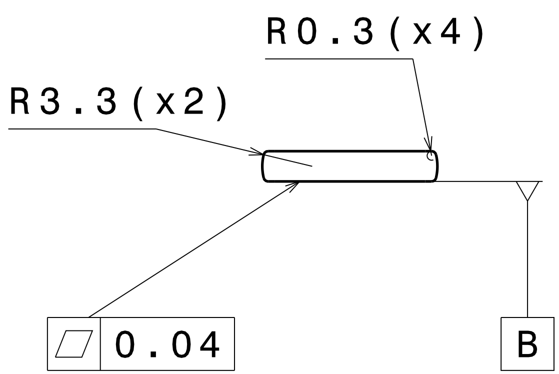
Variable Displacement Oil Pump

SCALE: 1:1 | WEIGHT (kg): | PArt: Driving Shaft | SHEET: 1/1

I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_

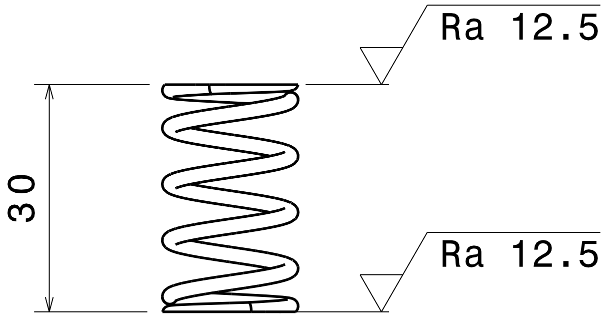


Top view
Scale: 2:1

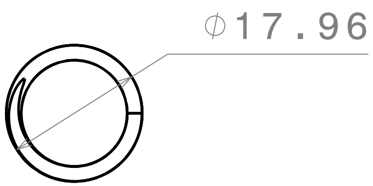


DESIGNED BY: Joao Batista DATE: 14-10-2014	MATERIAL: Steel 42CrMo4 GENERAL TOLERANCING: ISO 2768 - mk; ISO 8015	
SIZE A4		Variable Displacement Oil Pump
SCALE 1:1	WEIGHT (kg)	Part Vane — SHEET 1/1

Rev.	
I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_



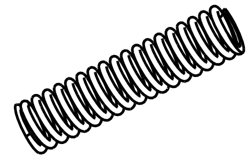
Front view
Scale: 1:1

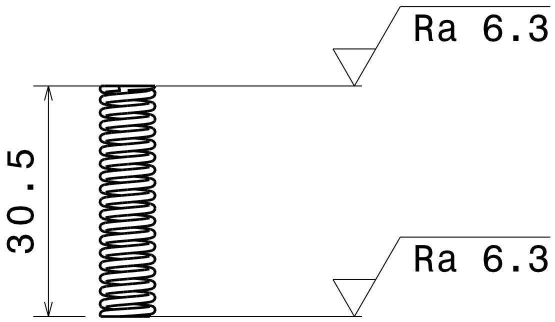


Top view
Scale: 1:1

- START OPENING SPRING VALUE=11.5mm
- WIRE DIAMETERr=2 +/- 0.025mm
- MEDIUM DIAMETER=16 +/- 0.1mm

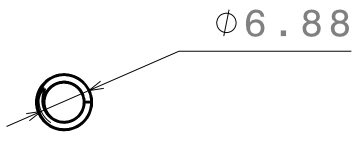
DESIGNED BY: Joao Batista DATE: 14-10-2014	MATERIAL: Steel en 10270-3 steel		I	_
			H	_
	GENERAL TOLERANCING: ISO 2768 - mk; ISO 8015		G	_
			F	_
SIZE A4		Variable Displacement Oil Pump	E	_
			D	_
SCALE 1:1	Part: Control Chamber Spring	SHEET 1/1	C	_
			B	_
			A	_





- START OPENING SRPING VALUE=19,9mm
- WIRE DIAMETER=1 +/-0.025 mm
- MEDIUM DIAMETER=7.30 +/-0.1mm

Front view
Scale: 1:1



Top view
Scale: 1:1

DESIGNED BY:
Joao Batista

DATE:
14-10-2014

MATERIAL: Steel en 10270-3 steel

GENERAL
TOLERANCING: ISO 2768 - mk; ISO 8015

SIZE
A4

Variable Displacement Oil Pump

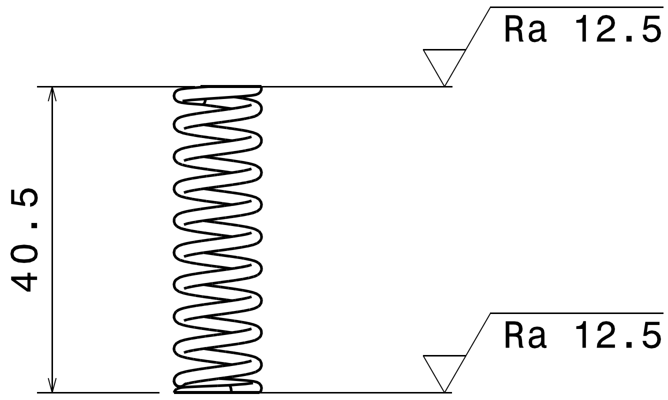
SCALE
1:1

WEIGHT (kg)

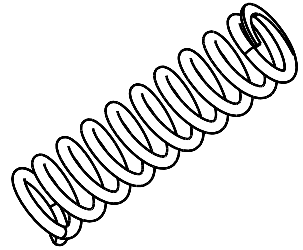
DRAWING NUMBER
Mec Control Valve Spring

SHEET
1/1

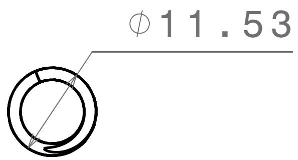
I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_



Front view
Scale: 1:1



- START OPENING SRPING VALUE=18.9
- WIRE DIAMETER=1.6 mm +/-0.025mm
- MEDIUM DIAMETER=11.6 +/-0.1mm



Top view
Scale: 1:1

DESIGNED BY: Joao Batista
DATE: 14-10-2014

MATERIAL: Steel en 10270-3 steel

GENERAL TOLERANCING: ISO 2768 - mk; ISO 8015

SIZE: A4

Variable Displacement Oil Pump

SCALE	WEIGHT (kg)	Part	SHEET
1:1		Safety Discharge Valve Spring	1/1

I	_
H	_
G	_
F	_
E	_
D	_
C	_
B	_
A	_

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